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OF

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

CONTAINING

THE PROCEEDINGS

SEPTEMBER 1908

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Gustav Herrmann

HONORARY MEMBER

OF

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CONTENTS

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HISTORY OF THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

Preliminary Report of the Committee on Society History

Chapter V

THE EUROPEAN TRIP OF 1889

144 As has already been noted, there was received by the Society, and presented at the Scranton meeting in October 1888, an invitation from Mr. E. N. Carbutt, President of the Institution of Mechanical Engineers, to such members of the Society as might be visiting Europe in connection with the Paris Exposition of 1889, to become the guests of the Institution of Civil Engineers, The Iron and Steel Institute, and the Institution of Mechanical Engineers.

145 As a result of a canvass of the membership of the Society, as well as of the American Institute of Mining Engineers and the American Society of Civil Engineers, it became evident that a large party desired to avail themselves of this opportunity. It was announced at the Erie meeting, in May 1889, that the plans for the trip were practically completed.

146 A very full report of the entire trip is included as an Appendix to volume ten of the Transactions of the Society, while an admirable review of the event formed a portion of the address of Mr. Henry R. Towne, President of the Society during that memorable year and himself one of those to whose efforts and activity much of the success of the trip was due.

Under the direction of the Council, the Committee on Society History has arranged to present the results of its investigations to the members of the Society.

The Preliminary Report will appear in the Proceedings of the Society from month to month, and thus enable the matter to be open to comment during its completion. It is especially desired that any member who may be in the possession of facts or information bearing upon the various points as they are thus made public will communicate with the committee, in order that the final and completed report may have the advantage of the collaboration of the membership at large.

147 It has been thought advisable to give here that portion of Mr. Towne's address relating to the trip, as the best condensed account possible, and the following pages are therefore reprinted from the presidential address of 1889.

During the year 1889 there took place an occurrence unprecedented in the annals of American engineering societies. A party consisting of 360, including their guests, went abroad as guests in the first instance of the Institution of Civil Engineers of Great Britain, and subsequently as guests of similar organizations in France and in Germany. It happened that the inception of the excursion came from the Institution of Mechanical Engineers of Great Britain and was extended to this Society as being its nearest sister on this side of the ocean. But as soon as the matter became noised in England, the Institution of Civil Engineers, which is there the parent society of all of their engineering associations, immediately claimed the right and privilege of becoming the hosts of the occasion, and of extending the invitations to each of the three or four engineering societies on this side. It followed from this that the formal invitation came directly to each of our American societies from the Institution of Civil Engineers. The most active part in promoting the project on this side was borne by this Society, but the other societies soon recognized the importance of the matter and all worked harmoniously together for the common result. It is to be hoped that the temporary union which thus arose may be the precursor of another, larger, and more permanent union among our American engineering societies.

The start was made on the 25th of May on the steamer City of Richmond, carrying 170 members of the party, consisting almost wholly of members of this Society and of the American Institute of Mining Engineers. That vessel had been chartered by our party expressly for the occasion and carried no other saloon passengers. The response to the invitation was so large, however, that the overflow, beyond what that steamer could accommodate, necessitated other arrangements, and so another party of 105 followed on the steamer City of New York on the 29th of May. There were still others who came in scattering groups, and some were already on the other side.

The voyage of the City of Richmond is a memory which all of us who had the privilege of taking part in it will ever recall with the greatest pleasure. It was harmonious from beginning to end. A committee was organized on the second day after sailing and had sessions every day of the voyage—indeed long sessions, as there was much work to be done in preparation for the affairs to be carried out on the other side, more than any of us had realized. The members of the party, both ladies and gentlemen, soon became well acquainted, and the voyage came to resemble a large yachting party rather than an ordinary trip across the Atlantic.

We reached Liverpool on Tuesday, June 4, and before we had set foot on English soil received a foretaste of English hospitality. There came out to meet us in the Mersey a tender carrying a committee of the local reception committee at Liverpool headed by Mr. Alfred Holt, their chairman (reputed to be the largest individual shipowner in the world), Mr. Daglish, Mr. West, and a number of other gentlemen. They boarded our ship and greeted us with words of hearty welcome, took charge of us during the landing, facilitating our passage of the Customs

authorities, and from that time until we left Liverpool were ceaseless in their endeavors for our comfort and enjoyment.

The City of New York arrived two days later, in the early morning, and with that day began the regular excursions which had been planned for our entertainment. Our hosts in England were the Institution of Civil Engineers. The individual members taking part in the entertainment, most of whom came expressly to Liverpool to greet us, were the president, Sir John Coode, Sir Frederick Bramwell, Sir Lothian Bell, Sir James Allport, Mr. Adamson, Sir Henry Bessemer, Sir Geo. Bruce, Mr. Cowper, and many others which I will not take time to name; but among them all no name made itself more familiar to us, or will ever be more warmly remembered, than that of the secretary, Mr. James Forrest.

It became necessary for the party, comprising, as it did, members of this Society, of the Civil Engineers, and of the Mining Engineers, together with a few members of the Electrical Engineers, to create some kind of an organization which should represent the united party during its travels in Europe. A joint committee was appointed to accomplish this purpose, and the result of their labors was the selection and recommendation of the following list of officers, who were unanimously elected by the joint party: Mr. Whittemore as honorary chairman, Mr. Henry R. Towne as chairman; and as officers or associates: Mr. Chanute, Mr. Woodbury, Mr. Clarke, Professor Hutton, Mr. Wiley, Mr. Dempster, Mr. Kent, Mr. Archbald, Mr. Baldwin, Mr. Fisher, Mr. Hawkins, Doctor Torrey, Mr. Bond, Mr. Forsyth, Mr. Oberlin Smith, and Mr. D'Invilliers. The treasurer was Mr. Hunt; the honorary secretary, Mr. Emery; the secretary, Mr. Kirchhoff. It is an evidence of the cleverness with which the nominating committee did its work, that out of the 21 names above, there are 13 who are members of the Society of Civil Engineers, 13 who are members of the Society of Mechanical Engineers, and 9 who are members of the Institute of Mining Engineers. The joint committee proved acceptable and accomplished its work satisfactorily, although the work proved to be much larger than would have been appreciated beforehand and demanded a great deal of time and care.

Our first full experience of English hospitality came at Liverpool in the form of a dinner given by Sir John Coode at Liverpool, the evening after the City of New York arrived, to a few of the officers of the joint party, followed during the evening by a conversazione at the Town Hall, given by the Mayor, Mr. Cookson, and attended by the whole of our party and a great number of ladies and gentlemen from Liverpool—a most brilliant assemblage.

The next morning we divided into two parties, one going to the Mersey Docks under the guidance of officers of the Dock Estate, who have charge of the most vast and expensive system of dock construction in the world, the extent of which is simply marvelous, and to us in America utterly unknown. The tides in Liverpool, and indeed all around the English coasts, average nearly thirty feet in height, entirely precluding the use of a wharf system such as we have here, and necessitating the putting of all vessels into docks closed by gates which are opened only for about an hour at high water.

The other party went through the Mersey Tunnel, a great work connecting Liverpool with Birkenhead, which has been recently completed under the guidance of Mr. Rowlandson, the engineer, and then to the Laird shipyards, where 576 vessels have been built within the last 30 years. They were entertained at a magnificent luncheon served in a tent on Mr. Laird's grounds, and then visited

one of the great steamers then being built for the Hamburg line—a sister ship of the Augusta Victoria,—and finally were brought back to Liverpool, arriving at the great landing stage which is used for tenders and ferryboats to deliver their passengers upon, said to be the greatest floating structure in the world, and having a total length of 2063 feet.

The next day the party divided again; one section going to Crewe, the location of the great constructive works of the London and Northwestern system, corresponding to Altoona on the Pennsylvania system, where they make steel rails, build locomotives, and conduct most of the mechanical operations of the line. The extent of those works is probably familiar to all of you, but it is interesting to note that the capital of that great corporation is \$528 000 000, with an annual revenue of \$51 000 000, and with 60 000 employees. It is also interesting to note that, even in that snug little island, one railway system can control and operate 2500 miles of line. The Crewe works cover 116 acres of land, of which 36 are under roof.

The other section of the party on that day went to Horwich, on the line of the Lancashire and Yorkshire Railway, and inspected a similar plant there, but one even more interesting than that at Crewe in this respect—that while Crewe has grown up almost from the commencement of railway operations in England and is to some extent a patchwork, although a vast and most highly organized one, the new plant of the Lancashire and Yorkshire Railway at Horwich is entirely new, has been built within the last three years, was laid out and organized by commencing with a clean sheet of paper and an unbroken piece of ground admirably chosen, and has a series of vast buildings designed harmoniously with reference to their intended uses and in the light of the best and latest modern experience, including that of Crewe. The mechanical engineer of that system, Mr. Aspinall, who has charge of the Horwich works, although a younger man than Mr. Webb, the presiding genius at Crewe, is his equal apparently in talent and organizing capacity, and, working as he does with this newer and more modern plant, is making a record which certainly will be a good second to that of Crewe. In the manufacturing department, where they make the smaller products, such for example as their switch and signal apparatus, Mr. Aspinall has introduced a great deal of American machinery and American methods of manufacture, and it seemed to me that the place compared favorably with any private establishment I have ever visited. These works cover 85 acres of ground, of which 13½ are under roof.

In the evening of that day the two parties united at Manchester, where a reception and banquet were tendered us at the Town Hall, presided over by the Mayor of Manchester and attended by a great many of the prominent citizens. It was a delightful occasion and even more elaborate than the reception at Liverpool.

The next day the party visited the great ship canal between Manchester and Liverpool, 35 miles in length, the contract price for which was \$28 000 000, on which 15 000 men were employed.

The next week being the Whitsuntide Holiday or recess, was utilized for excursions not connected with the engineering part of our visit. The party broke up into two groups, one going through North Wales, the other through the Midland counties, reuniting in London. It is fitting to remark at this place that all through the trip the courtesies extended to the American engineers by English

railway officials were marked and generous to the greatest extent. The London and Northwestern system gave free transportation from Liverpool to London, including a return privilege at whatever time the holder of the ticket desired, and other systems followed later when the party had reached London and made excursions from that point.

On Thursday, the 13th of June, those wonderful eight days of hospitality in London began with a choral service in Westminster Abbey, conducted by Dean Bradley, who gave an address of welcome to our American party; then a brief visit to the Houses of Parliament and in the afternoon a reception by the Institution of Civil Engineers. The latter was opened by an address of welcome from Sir John Coode, the president, the words of which have been beautifully illuminated and framed and presented to this Society, and also to our sister societies here, by the Institution of Civil Engineers, and a copy hung in our new rooms. Our party was especially fortunate in having with it at that time one of our oldest and most honored members, to whom was committed the duty of replying to the address of welcome from Sir John Coode, and who did it in a manner which more than fulfilled our expectations. Professor Thurston's admirable address on that occasion was one for which all of our party felt grateful and of which all of us were proud.

In the evening of that day a dinner was given to the party by the Institution of Civil Engineers in the old and historic Guildhall of London, a building which we were told had never before within memory been used for any purpose not directly connected with the civic hospitality of the City of London. It was a great compliment. The dinner was elegant beyond easy expression and was dignified and notable in every particular. Among the guests of the occasion were the American Minister, Mr. Robert Lincoln, Sir Edward Thornton, Lord Armstrong, Archdeacon Farrar, Dean Bradley, Sir Henry Bessemer, Sir William Thompson, Mr. Latimer Clarke, Sir James Douglass, Mr. Gilchrist, Mr. Mather, Sir E. J. Reed, Professor Unwin, and a great many others whose names are familiar on this side of the Atlantic as well as on the other. One of the pleasing incidents of the evening was the address given by Mr. Lincoln, which was worthy of the occasion and able throughout, and at the close of which he gave utterance to a sentiment especially complimentary to us and typical of the character of our times and of the change in sentiment which is taking place in the world. Addressing our united party of engineers, English and American, he said that "engineers throughout the world are doing more than any other agency at the present time to bring about the brotherhood of the nations, and to render superfluous such offices as that which I now have the honor of holding."

The next day was devoted to visits to the docks and gasworks, to drainage works, to the great Tower bridge across the Thames, to Greenwich, to the Yarrow shipyard, to various engineering works, and by a fraction of the party, to a visit to Lambeth Palace, where the Archbishop of Canterbury received the guests and conducted them personally through the edifice. On the following day, June 15, we were taken by special train over the Great Western Railway to Windsor, where the Queen had given special permission for our party to go through the palace, and to see not only those parts which are usually open to the public but also the private apartments, which were exceedingly interesting. A small fraction of the party went on that day to the grounds of Mr. Boulton, at Totteridge, where they witnessed a remarkable presentation of the Midsummer Night's

Dream, given in the open air. The evening of the day concluded with a reception tendered to our party by Lord Brassey at his beautiful house in London, where we saw many of the wonderful curios collected by himself and the late Lady Brassey during their yachting tours around the world. The following day was a Sunday, and on the next day, Monday, the party went in the morning to see the Royal Palaces in London—St. James and Buckingham. It was one of the coincidences of this visit that we were greeted there with the strains of Yankee Doodle and Hail Columbia, the day being, as one of our party recalled, the anniversary of the battle of Bunker Hill. On the afternoon of this day Lady Burdett-Coutts gave a garden party and reception at her London residence. The following day was devoted to a trip to water works and pumping plants, to Hampton Court Palace and Bushy Park, and the day following to similar visits to railway stations and the great plant of the London Electric Supply Corporation, the ladies going to the flower show of the Royal Botanic Society; and a party of our members, unfortunately a small one, able to avail themselves of the privilege, spent the day in a visit to the residence of Professor Tyndall, who had invited as large a party as his house was capable of entertaining. Those who went received at his hands, I am told, a most cordial and delightful reception, and it is pleasing to mention a fact which I also learned from those who were fortunate enough to be there, that the response made over the luncheon table to the remarks of Professor Tyndall by our honorary chairman, Mr. Whittemore, were eloquent and beautifully fitting to the occasion. Other hospitalities were extended to individual members of the party on occasions which did not admit of their being made general. One or two of the London clubs gave admission to our members, as had also been done in Liverpool, and in every way the hospitality of our English cousins was cordial beyond any mere words of expression. I think that all of our party, in undergoing these experiences, realized that while there was of course a large amount of personal hospitality underlying it, and still more of professional welcome, the true motive prompting these manifestations from our English friends was that of deep and sincere cordiality towards America and Americans. This was made evident to us throughout the whole of our English experience, and it struck me that the feeling of *kinship* on the part of the English toward us is even greater at the present time than the corresponding feeling which we entertain toward them. We Americans look back to England as the mother country and as such have for it the warmest feeling of affection, but on the part of Englishmen there can be of course no corresponding feeling toward this country. The fact of the kinship of the two peoples, however, I believe is more real to Englishmen at the present time than it is even to us, and I think that they realize more clearly than we the fact that together we constitute the two branches of the great Anglo-Saxon, English-speaking race, which in accomplishment, especially in the industrial world, is at present easily the leader among the nations in the march of civilization.

On June 20 our English friends again put us on a magnificent special train, and many of them accompanied us on it via the London, Chatham and Dover Railway, to Dover, and from there by a special steamer across the English Channel to Calais. Our crossing was on a beautiful sunny day, with bright sparkling water, and with no cause for discomfort.

Upon landing on French soil we again had an immediate greeting of hospitality from our new hosts, represented by members of the French Society of Civil

Engineers, who had come from Paris for the purpose. Again we were placed on a special train, tendered by the Northern Railway of France, and taken to St. Omer and Fontinettes, to see a new and unusual canal lift which had just been completed there, and thence on to Paris. Our hosts during our French visit were composed almost entirely of members of the French Society of Civil Engineers, headed by M. Eiffel, the president this year, M. Brull, a past president, M. Contamin, principal engineer of the wonderful machinery palace at the exhibition, which was awarded, I believe, the prize of 20 000 fr. tendered by an American for that feature of the exhibition which, in the opinion of a special committee appointed to make the award, represented the highest accomplishment and greatest usefulness. The committee's award was to the designers and builders of the wonderful machinery hall, a building having a span of 330 ft. and a length of about 1500 ft. The other members of the reception committee were M. Jousselin, M. Banderali, M. Pontzen, M. Alphand, who is the director-general of the Exhibition, M. Garnier, the world-famous architect, M. Haton de la Goupillière, who is the head of the École des Mines in Paris, M. Gottschalk, M. Charton, and many others. A few members of the joint committee were privileged to be the guests at a small but most delightful dinner given at one of the restaurants in the exhibition grounds by a gentleman whose name has been too little associated with this wonderful excursion, Mr. James Dredge, of London, the editor of the journal Engineering, and one of the leading representatives of the English section of the late Paris Exhibition. All of our societies were indebted more to Mr. Dredge than to any other one person for inaugurating the excursion, for enlisting English and French interest in it, and for contributing to the success of the whole undertaking. One member of our party, the treasurer of this Society, Mr. Wiley, knows the facts, but they are not yet fully appreciated even by the members of our party abroad; and I repeat that no amount of thanks which we can express or tender to Mr. Dredge would cancel the obligation which we owe him.

On Saturday, the 22d of June, our party went to the exhibition under the conduct of members of the French Society, and were taken through a portion of it, and then to the Eiffel Tower, after ascending which we were entertained at a luncheon on the lower platform of the Tower, presided over by M. Eiffel, the president of the French Society, and attended very numerously by members of the French Society and guests, including Mr. Whitelaw Reid, the American Minister, and General Franklin, our commissioner to the exhibition.

Our stay in Paris included many other visits—to the great sewers, to the Gobelin Tapestry Works, to M. Pasteur's laboratory, to the École des Mines, to the great omnibus and cab companies, to the sewerage and pumping stations, to Sèvres Porcelain Works, and so on. The social features of our entertainment in Paris included, besides what I have already mentioned, receptions to a part or the whole of our joint party by President Carnot, by the Prefect of the Seine, and by the Municipal Council. It was a pleasant feature of our reception at the latter place that one of the speakers on our side, Professor de Garay of the City of Mexico, a member of the American Society of Civil Engineers and an accomplished scientist and gentleman, responded most eloquently in the French language, as was done by other members of our party on other occasions. The Institution of Mechanical Engineers, which had been the first to extend an invitation to us to visit England, happened to have their summer meeting in Paris just at

the close of our stay there, and extended to those of our members who remained an invitation to their dinner and to their sessions, so that English hospitality followed us even on French soil. Then came the disbanding of the party, some returning home, others going south, and a considerable number going into Germany, where we afterward heard of them as receiving hospitality even more overwhelming than that which had greeted us either in England or in France. Still others of us came back to London, and a very small number, seven only being obtainable, were privileged to take part in a small but unique entertainment given by Mr. Dredge, again our host, in order, as he supposed, to enable us to present a handsome silver loving cup to Mr. James Forrest, as a token of appreciation from the members of the party to him, for what he had done for us during our visit abroad. The committee having the matter in charge, however, appreciated that Mr. Dredge was entitled to a loving-cup as well as Mr. Forrest, and two cups were prepared, each suitably inscribed. Each of the two recipients knew that the other was to be presented with a cup, but neither knew that he was to receive one himself, and there was a very pleasant and amusing dénouement when the second cup came out.

On July 22 a number of our party again came together and for the last time accepted the hospitality of the Midland Railway and a special car to Derby, where they were the guests of Mr. John Noble, the general manager of the company, and several of their directors. After a handsome luncheon in the directors' room, we visited their works, which are similar to those at Crewe although not quite so large, and ended the day by reaching Liverpool in preparation for the homeward voyage. The party which reassembled in this way at Liverpool, numbering more than fifty, came home together on the City of Paris, reaching New York just three minutes too late to break the ocean record. The rest of the party came home in scattering groups, but more than fifty of us came together once more in this city on the 25th of August to be the recipients of hospitalities at the hands of some of our friends at home, at the Engineers Club, where a handsome dinner was given to the returning guests, a proceeding likened to heaping coals of fire on our heads, the hosts being those who had not been able to participate in the pleasures which we had just returned from enjoying. And so ended an excursion remarkable in every sense of the word, absolutely unique and without precedent, and one which will ever be a delightful memory to those whose privilege it was to take part in it.

PERFORMANCE OF BELT CONVEYORS

By Edwin J. Haddock, Columbus, Ohio

Member of the Society

Of the half dozen forms of conveyors commonly used for handling material in bulk, or for continuous conveying, one of well deserved and increasing popularity on account of its capacity, economy of power, noiselessness, and gentle handling of its load, is the belt conveyor.

2 This form of conveyor is a very efficient means of conveying coal, crushed ores, grain, small goods in packages, etc., when the transportation is in a straight line, either horizontal, or inclined at an angle not exceeding, in good practice, 20 deg.

3 Some experiments with belt conveyors, made some time ago by the company with whom the writer is connected, may be of interest. Data relating to the following points were desired:

a Other conditions remaining the same, what will be the effect on the traction of changing the diameter of the driving pulley?

b What is the effect of different arcs of contact on the tractive force exerted by the driving pulley?

c What is the effect of different initial tensions of the belt on the tractive force?

d What is the value of rubber covered pulleys as compared with plain ones?

4 An experimental belt conveyor was constructed as shown by Fig. 1 and Fig. 2. This conveyor was driven by a 220 volt d.c. motor, and the driving machinery was calibrated for torque as shown in Fig. 3 and Fig. 4.

Presented at the Detroit Meeting (June 1908) of The American Society of Mechanical Engineers. This paper was received too late to be included in June Proceedings.

5 A prony brake was placed on the head shaft of the conveyor (the conveyor belt having been removed) and a pair of platform scales

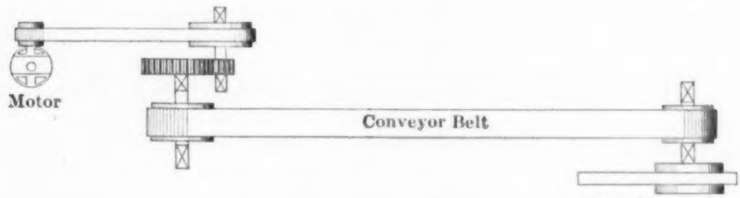


FIG. 1

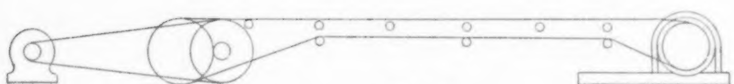


FIG. 2

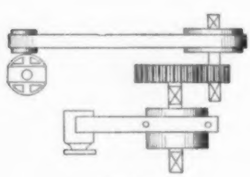


FIG. 3

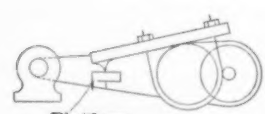


FIG. 4

placed under it, as shown in the illustrations. A d.c. voltmeter and ammeter were placed in the electric circuit, and the readings, shown in Table 1, taken, from which the calibration curve, Fig. 5, was constructed, by which the torque on the head shaft may be determined from any reading of the ammeter, the voltage being kept at 220.

TABLE I CALIBRATION OF DRIVING GEAR

Scales Load	Torque 12 in. Radius	Amperes, average four observations	Volts, average four observations	Watts average four observations	Amperes at 220 volts
0	0	5.0	222.0	1110	5.05
25	225	9.87	225.7	2229	10.13
50	450	14.25	222.5	3170	14.41
75	675	18.87	220.0	4152	18.87
100	900	22.75	218.0	4960	22.54
125	1125	27.25	216.5	5898	26.81
150	1350	32.62	218.0	7112	32.34
175	1575	37.50	213.0	7996	36.35
200	1800	42.00	214.0	8988	40.85

6 A 12 in. 4-ply conveyor belt, 158 ft. long, was placed on the conveyor, a joint being made by lapping and riveting the ends. The total weight of the belt was 380 lb., or 2.405 lb. per foot. The retarding brake was placed at the foot shaft of the conveyor.

7 Experiment 1 was made to determine the effect of changing the diameter of the driving pulley. The tension was kept constant by

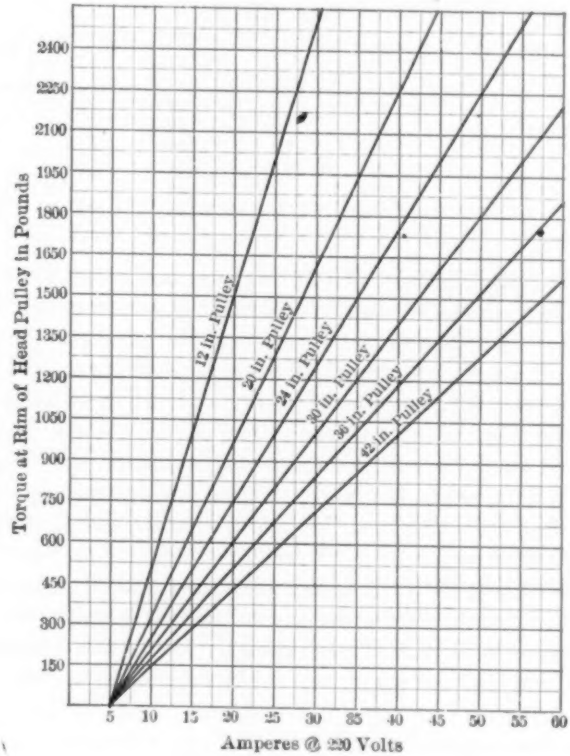


Fig. 5

EXPERIMENT 1

measuring the sag of the belt when the conveyor was idle. This measurement was made both before and after each test, care being taken to have it uniform for all the tests.

8 In all the tests the brake on the foot shaft was tightened until the belt slipped on the head pulley, and the reading was taken at that

moment. An average of at least four readings is given in the table. In each case, the ammeter read one to three amperes higher just before the belt slipped than it did as the belt was slipping, the hand dropping back to the reading again.

9 The observations shown in Table 2 show the effect of different diameters of the driving pulley.

TABLE 2

Order of test	Amperes at 220 v.	Diameter of pulley in inches	Belt slipped at lb.
4	12	12	720
3	19	20	855
2	22	24	855
1	27	30	895
5	32	36	910
6	35½	42	885

10 The initial tension of the belt was not measured in pounds, care only being taken to keep it uniform as above described.

11 The belt was new and covered with the usual preservative mineral powder, which was wiped off from time to time. This circumstance, together with any possible errors of tension, would account for most of the discrepancies shown. The results clearly show that the diameter of the pulleys from 42 in. to 20 in. had no appreciable effect on the belt in question (4-ply), but that a decided falling off occurred in the 12 in., indicating that this pulley was too small to run with a belt of this thickness.

12 From these data the writer assumes that if a pulley is not less than 5 in. in diameter for every ply of the belt it will exert its maximum tractive force, the excess diameter above this being simply a matter of convenience for drive. If the diameter is reduced much below this amount, the belt will not adhere closely to the pulley, and therefore the complete benefit of the pressure will not be realized.

EXPERIMENT 2

13 The second experiment was made to determine the effect of different arcs of contact. The conveyor was arranged as in Experiment 1, but considerable difficulty was met with in getting the tension uniform, and very erratic results were obtained.

14 The foot shaft was then rearranged, as shown in Fig. 6, the drive machinery at the head of the conveyor being used without any

change, except that the snubbing idlers were modified to get the required arc of contact.

15 With a 42 in. pulley on the head shaft, a 1000 lb. tension weight at the foot shaft, the following results were obtained, as shown in Table 3.

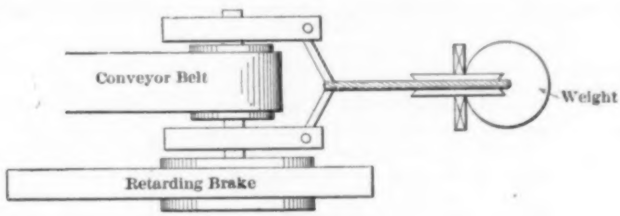


Fig. 6

TABLE 3 EFFECT OF DIFFERENT ARCS OF CONTACT

Arcs of contact	Amperes	Effective pull on belt in pounds	Load 180 deg. = 100
1/6 or 60 deg.	11	180	32
1/3 or 120 deg.	19	405	73
1/2 or 180 deg.	24	550	100
5/8 or 225 deg.	32	775	141
2/3 or 240 deg	34	840	153

16 From this table, the curve shown in Fig. 7 was constructed. It will be noticed that below the 180 deg. contact, the curve is fairly a straight line, but above this point there seems to be a curve in which the traction increases at a more rapid rate. It is unfortunate that more observations were not taken from the 180 deg. line up to get the exact form of this curve. Making the curve a straight line, however, throws the error on the safe side.

EXPERIMENT 3

17 Experiment 3 was made to determine the effect of initial tension on the tractive force: With the same arrangement of conveyor as in Experiment 2, including the 42 in. pulley on the head shaft, and the snubbing idler arranged to give a contact of 180 deg. on the drive pulley, different tension weights were hung on the foot of the conveyor and the readings shown in Table 4 were obtained.

18 The tests are given in the order made. When the 2000 lb. weight was applied, the belt stretched abnormally from its length

under the 1500 lb. weight, the additional 500 lb. causing the belt to elongate about 3 ft. in the total length of 158 ft. The belt then seemed to stop stretching, and remained at a constant length until the load was removed. As the weight was reduced gradually to 750 lb., the belt recovered 15 in. of this stretch, while the weight of 3000 lb. caused the belt to stretch 8 in. longer than the 2000 lb. weight had

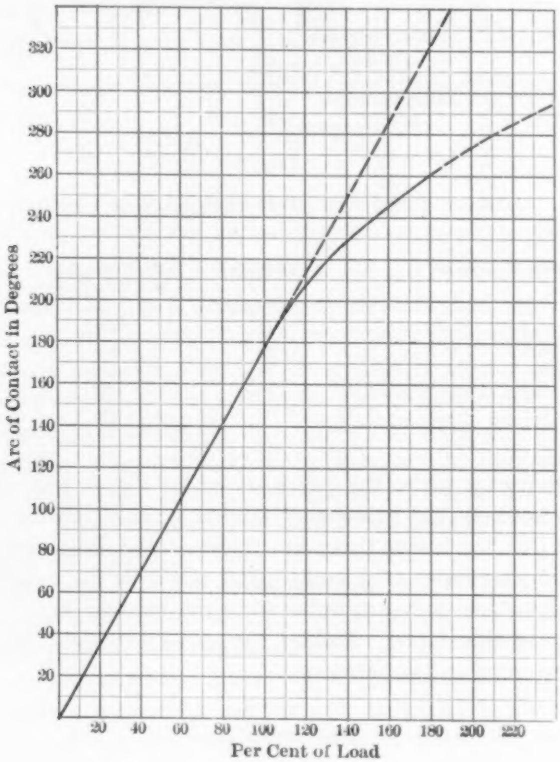


Fig. 7

done. The belt was then run under this tension for a period of 30 minutes without changing its length to any appreciable degree. The belt strain of 1500 lb., caused by the application of a 3000 lb. weight, is equivalent to a stress of 31.25 lb. per inch per ply. No reason appears against using 40 lb. per inch width per ply (the manufacturers' guarantee) as to the ultimate working strength of any rubber con-

veyor belt. This is the maximum value which should be allowed in the general equations following.

19 From Table 4 Fig. 8 was constructed, which shows the various belt strains when the arc of contact is 180 deg., and the initial tension on the belt is known.

TABLE 4
EFFECT OF DIFFERENT TENSIONS ON THE SAME PULLEY

Tension Weight	Initial belt tension	Amperes	Effective pull on belt	BELT STRAIN	
				Maximum	Minimum
650	325	14.0	260	455	195
800	400	16.5	330	565	235
1300	650	21.0	550	925	375
1500	750	29.0	690	1095	405
2000	1000	38.0	950	1475	525
1500	750	31.0	750	1125	375
1250	625	25.0	575	913	337
1000	500	22.0	490	745	255
750	375	18.0	370	560	190
3000	1500	56.0	1450	2225	775

EXPERIMENT 4

20 Experiment 4 was made to determine the relative value of rubber faced vs. iron faced pulleys. A 24 in. rubber covered pulley was compared with a 24 in. iron faced pulley, on the experimental conveyor, with 180 deg. arc of contact, and a tension weight of 2000 lb. throwing an initial strain of 1000 lb. on the belt, the results shown in Table 5 were obtained, or 7 per cent in favor of the rubber faced pulley when the belt was clean and dry.

TABLE 5 EFFECT OF RUBBER FACING ON PULLEYS

	Amperes	Traction pounds	Per cent
24 in. iron faced pulleys....................	24	960	100.0
24 in. rubber faced pulleys................	25½	1035	107.8

21 This experiment was performed at two different times, with an interval of several weeks between, and with substantially the same results. In order to carry it a little further, the device shown in Fig. 9 was provided to determine the effect of different conditions of the belt surface on rubber and iron faced pulleys.

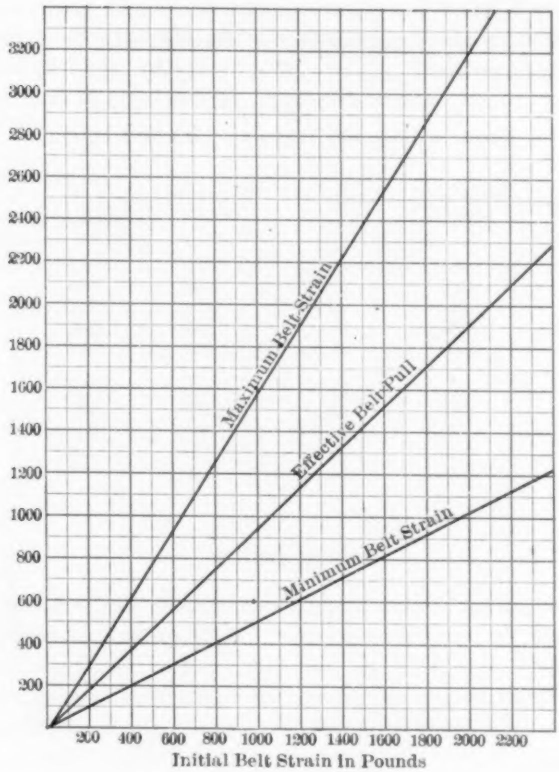

3200
3000
2800
2600
2400
2200
2000
1800
1600
1400
1200
1000
800
600
400
200
Maximum Belt Strain
Effective Belt Pull
Minimum Belt Strain
200 400 600 800 1000 1200 1400 1600 1800 2000 2200
Initial Belt Strain in Pounds

FIG. 8

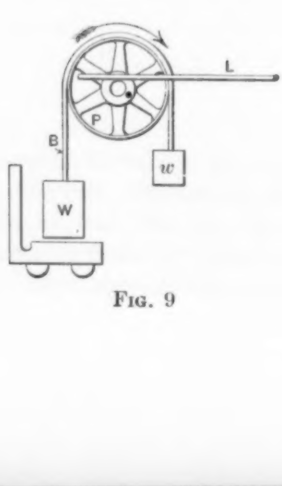

L
P
B
w
W

FIG. 9

22 In this figure, W represents a heavy weight on the platform scales; B is a 6 in. 3-ply rubber belt; P is a 24 in. iron or rubber faced pulley; w is a small tension weight; and L is a hand lever by which the pulley was rotated. When the pulley was rotated slowly by hand, in the direction of the arrow, the scales would indicate the amount of weight not lifted by the traction of the belt, and hence the belt traction could be computed. The results are shown in Tables 6, 7 and 8, which also show the effect of foreign material between the belt and pulley.

23 When foreign material was introduced between the belt and the pulley it seemed to crush in between the two surfaces, so that if it was of a gritty or abrasive nature, it increased the tractive force, while a material which acted more or less as a lubricant, diminished that force. The abrasive material scratched the belt very much, and gave evidence of shortening its life.

24 From the foregoing experiments, the following general equations for traction may be deduced:

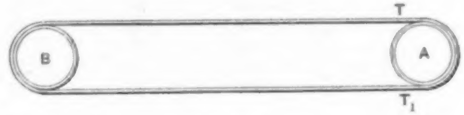


FIG. 10

THEORETICAL BELT CONVEYOR

Let Fig. 10 represent such a conveyor

A, the driving or head pulley
B, the driven or foot pulley
T, the maximum pull on belt
T_1, the minimum pull on belt
F, the tractive force exerted by the pulley A
T_2, the initial tension of belt
i, the portion of the pulley in contact with the belt

$$= \frac{\text{arc of contact in degrees}}{360}$$

25 c is a constant depending on the arc of contact $= 2$ from 0 to 180 deg., and about 2.25 at 240 deg. For practical purposes, c may be taken at 2 for all arcs and the errors will be on the safe side.

26 k is a constant depending on the condition of the belt $= 1$ for good dry belts, but will vary from 40 per cent to 200 per cent of this value under different conditions. (See Tables 6, 7 and 8.)

TABLE 6 IRON FACED PULLEY

	Scales load $W = 227$	Net weight lifted lb.	Counter weight w lb.	Traction lb.
Clean dry belt	22	205	52	153
Clean damp belt	34	193	52	141
Dry coal dust	33	194	52	142
Damp coal dust	58	169	52	117
Dry clay	78	149	52	97
Damp clay	90	137	52	85
Dry slate	79	148	52	96
Damp slate	45	132	52	130
Dry sharp sand	110	117	52	65
Damp sharp sand	80	147	52	95

TABLE 7 RUBBER FACED PULLEY

	Scales load $W = 327$	Net weight lifted lb.	Counter weight w lb.	Traction lb.
Clean dry belt	109	218	52	166
Clean damp belt	120	207	52	155
Dry coal dust	152	175	52	123
Damp coal dust	195	132	52	80
Dry clay	190	137	52	85
Damp clay	175	152	52	100
Dry slate	63	264	52	212
Damp slate	50	277	52	225
Dry sharp sand	152	175	52	123
Damp sharp sand	152	175	52	123

TABLE 8
TRACTION OF IRON FACED VS. RUBBER FACED PULLEY

	Iron faced lb.	Rubber faced lb.	Iron faced per cent	Rubber faced per cent	Ratio Iron = 100
Clean dry belt	153	166	100	108	108
Clean damp belt	141	155	92	101	110
Dry coal dust	142	123	92	80	86
Damp coal dust	117	80	76	52	68
Dry clay	97	85	63	55	88
Damp clay	85	100	55	65	117
Dry slate	96	212	68	138	221
Damp slate	130	225	85	147	173
Dry sharp sand	65	123	42	80	190
Damp sharp sand	95	123	62	80	129

27 The general formulae for traction may be represented thus:

$$(a)\ F = i\,c\,k\,T_2 = T - T_1 = \frac{i\,c\,k}{2}(T + T_1) = \frac{2\,i\,c\,k\,T}{2 + i\,c\,k} = \frac{2\,i\,c\,k\,T_1}{2 - i\,c\,k}$$

$$(b)\ T_2 = \frac{T + T_1}{2} = \frac{F}{i\,c\,k} = \frac{2\,T}{2 + i\,c\,k} = \frac{2\,T_1}{2 - i\,c\,k}$$

$$(c)\ T = T_1\left(\frac{2 + i\,c\,k}{2 - i\,c\,k}\right) = T_2\left(\frac{2 + i\,c\,k}{2}\right) = F\left(\frac{2 + i\,c\,k}{2\,i\,c\,k}\right)$$

$$(d)\ T_1 = T\left(\frac{2 - i\,c\,k}{2 + i\,c\,k}\right) = T_2\left(\frac{2 - i\,c\,k}{2}\right) = F\left(\frac{2 - i\,c\,k}{2\,i\,c\,k}\right)$$

27 Calling $c = 2$ and $k = 1$, these formulae reduce to the following for clean dry rubber belts on iron pulleys:

$$(e)\ F = 2\,i\,T_2 = T - T_1 = i\,(T + T_1) = \frac{2\,i\,T}{1 + i} = \frac{2\,i\,T_1}{1 - i}$$

$$(f)\ T_2 = \frac{T + T_1}{2} = \frac{F}{2\,i} = \frac{T}{1 + i} = \frac{T_1}{1 - i}$$

$$(g)\ T = T_1\left(\frac{1 + i}{1 - i}\right) = T_2\,(1 + i) = F\left(\frac{1 + i}{2\,i}\right)$$

$$(h)\ T_1 = T\left(\frac{1 - i}{1 + i}\right) = T_2\,(1 - i) = F\left(\frac{1 - i}{2\,i}\right)$$

28 It should be borne in mind that T indicates the maximum belt strain, and is seldom, if ever, properly represented by F, which is the pull or tractive force, and is less than T.

29 The makers of conveyor belts will guarantee a working value of 40 lb. per inch per ply for T in rubber belting, and this should not be exceeded. Good practice would indicate a value of 30 lb. or less per inch per ply.

30 When multiple drive heads are used, the traction of each drive pulley should be computed separately, starting with the maximum belt strain T for the first pulley, computing T_1, and making T for the second pulley equal T_1 of the first, and so on, for the number of pulleys, the difference between the first tension and the last one will be the tractive force.

31 The following tables have been computed from the foregoing formulae, with the values as indicated by these experiments, for clean dry rubber belts on iron faced pulleys. Due allowance must be made for any other condition of belts. See Tables 6, 7 and 8.

TABLE 9

DATA CONCERNING IRON PULLEYS FOR DIFFERENT ARCS OF CONTACT FOR CLEAN DRY RUBBER BELTS WITH A MAX. BELT STRAIN OF 30 LB. PER IN. PER PLY

Degrees contact	Belt strain pounds		Initial tension T_2	Effective pull or traction F
	Max. T	Min. T_1		
100	30	17.0	23.5	13.0
110	30	16.0	23.0	14.0
120	30	15.0	22.5	15.0
130	30	14.3	22.2	15.9
140	30	13.2	21.6	16.8
150	30	12.3	21.2	17.7
160	30	11.5	20.8	18.5
170	30	10.7	20.4	19.3
180	30	10.0	20.0	20.0
190	30	9.3	19.7	20.7
200	30	8.6	19.3	21.4
210	30	7.9	19.0	22.1
220	30	7.2	18.6	22.8
230	30	6.6	18.3	23.4
240	30	6.0	16.0	24.0
250	30	5.4	17.7	24.6
260	30	4.8	17.4	25.2
270	30	4.3	17.2	25.7
280	30	3.8	16.9	26.2

32 The carrying capacity of belts is largely a question of condition of material. On the whole, a belt will carry a larger number of cubic feet of fine material, such as grain or crushed coal, or rock, than it will if the material consists largely of lumps, such as Run of Mine coal. In the following table of capacity, such fine material alone is considered, carried on a horizontal conveyor.

33 The horse power required to drive belt conveyors is too broad a subject to be fully covered in this paper. It varies so much with the design and spacing of the idlers, and the local conditions, that no accurate rules applicable to all cases can be given. The following formula may be found useful for the average case:

W = total weight of material delivered per minute
W^1 = total weight of belt
S = speed of belt in feet per minute
h.p. = horse power
W_2 = maximum weight of material on belt at any one time.
H = total height to which material is elevated by conveyor

$$\text{h.p.} = \frac{(0.15\,W_1 + 0.07\,W_2)\,S + HW}{33\,000}$$

34 This will give the average h.p. In addition an allowance of from 4 per cent to 6 per cent should be made for each tripper, and all other unusual elements taken into consideration.

35 It is often desirable to change a belt conveyor at one or more points in its length, from horizontal to inclined, or to a greater incline by means of an upward curve, and it then becomes essential to know the curve assumed by an empty belt under such conditions in order to ascertain the minimum radius to which such a curve may be laid out, and the belt lay down on the idlers.

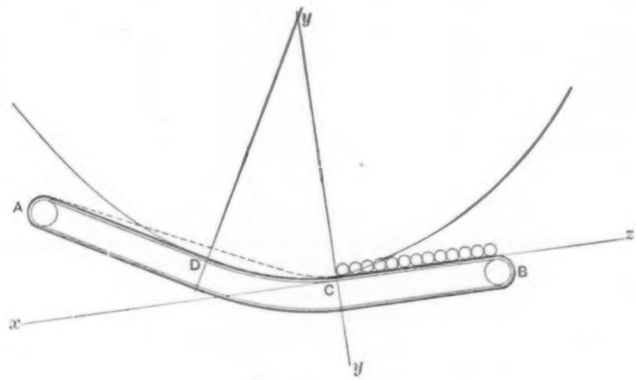


Fig. 11

Let Fig. 11 represent such a condition in which
A = the head pulley
B = the tail pulley
C = the point where curvature begins
D = the point where curvature ends
XZ and YY = Rectangular coördinates with origin at C
T = Maximum pull on belt at point C
W = Weight of belt per foot of length

TABLE 10

DATA CONCERNING IRON PULLEYS FOR DIFFERENT ARCS OF CONTACT FOR CLEAN DRY RUBBER BELTS WITH MAXIMUM BELT STRAIN OF 100 PER CENT PER INCH PLY

Degrees contact	Belt strain per cent		Initial tension T_2 in per cent	Effective pull or Traction F in per cent
	Max. T in per cent	Min. T_1 in per cent		
100	100	57	79	43
110	100	53	77	47
120	100	50	75	50
130	100	47	73	53
140	100	44	72	56
150	100	41	70	59
160	100	38	69	62
170	100	36	68	64
180	100	33	67	67
190	100	31	66	69
200	100	29	65	71
210	100	26	63	74
220	100	24	62	76
230	100	22	61	78
240	100	20	60	80
250	100	18	59	82
260	100	16	58	84
270	100	14	57	86
280	100	13	56	87

TABLE 11

SHOWING CROSS SECTION IN SQUARE FEET OF THE EFFECTIVE STREAM OF MATERIAL WHICH MAY SAFELY BE CARRIED BY BELT CONVEYORS

Width of Belt	Effective Cross section		Width of Belt	Effective Cross Section	
	Flat Belt	Troughed Belt		Flat Belt	Troughed Belt
10	0.014	0.05	26	0.19	0.40
12	0.025	0.07	28	0.23	0.47
14	0.04	0.11	30	0.27	0.55
16	0.057	0.15	32	0.32	0.63
18	0.078	0.19	34	0.36	0.71
20	0.102	0.23	36	0.40	0.80
22	0.13	0.28	42	0.57	1.00
24	0.16	0.34	48	0.77	1.20

36 It is evident that the tendency of the belt to rise above the idlers on the curve as shown by dotted line, will be the greatest when the strain at C is greatest, and when the belt from C to D is lightest, that is, the tendency to rise will be greatest when the less inclined portion of the conveyor is loaded up to the point C and empty from C to D. Then according to the law of the catenary, the hanging belt will assume the curve of a parabola whose equation is

$$X^2 = \frac{2T}{W} Y$$

from which the curve may be plotted.

37 The belt will under the above maximum conditions never fall within the curve as computed from the above formula, between the points C and D, so that if the idlers are brought up to or a little above this curve, the belt will never leave the idlers.

38 When conditions permit, the radius of the curve C to D is usually made 300 ft. which is generally found satisfactory for all belts.

GAS POWER SECTION

PROGRESS REPORT OF THE COMMITTEE ON STANDARDIZATION

In order to further the work of the Standardization Committee of the Gas Power Section, the chairman directed some important communications which had been received by the committee to be read before the Section at the session at Detroit on June 25, and these communications are here presented in the hope that they may elicit further discussion from the membership.

Communication from Mr. J. R. Bibbins

2 Mr. J. R. Bibbins submitted the following memorandum, this following along the lines of the paper presented by Dr. Lucke at the February meeting of the Section.

ENGINE CAPACITY

3 In view of the uncertainty attending the indication of small high speed gas engines, brake horse power should be used for the present at least in all expressions of engine capacity. Although it is perfectly feasible to indicate large slow speed engines with reasonable accuracy, basing the rating on mean effective pressure, it is difficult to draw the line where the use of the indicator should cease and power measurement in brake horse power begin. To be sure, on all large engines, brake horse power involves in the majority of cases, a determination of the generator efficiency. If it is not desired to employ data from builders' tests (which are always made on large machines by responsible builders) the only alternative is to make a test in the field, which is not a very difficult matter as compared to the rigging necessary for a brake test. In figuring over the generator data of the Norton test,[1] I was surprised to find that a considerable range in voltage had relatively little effect upon the efficiency curve, which would seem to indi-

[1] Published in Mid-November (1907) Proceedings.

cate that the inaccuracies incurred in electrical efficiencies are perhaps no greater than in indicating. In all tests of large engines, I would certainly advise that both indicator and electrical readings be taken, if for no other reason than to check the accuracy of results. These should be taken simultaneously by separate observers and by signals. The accuracy of the graphical method of plotting these observations has already been demonstrated in the Norton test.

TIME ELEMENT

4 Capacity should be expressed on a basis of *continuous operation* for full load rating and for a definite number of hours at maximum overload. If a builder will guarantee maximum overload continuously, so much the better, but this is hardly to be expected, and would not be representative of commercial conditions.

OVER-LOAD CAPACITY

5 Conservative builders have consistently held down their overload rating to provide for operative contingencies *entirely beyond their control,* e. g., weak gas. This is not just to the engine builder alone and applies only to the contract for an entire producer engine equipment. To protect him, the range of fluctuation or at least the lower limit in heat value of gas, should be definitely stated, also the composition of the gas, as later noted.

ALTITUDE

6 Although the customer is only interested in the capacity an engine will develop at his particular altitude, it is a fact that builders rating at sea level have an appreciable advantage over those rating at say 1000 ft. (the latter to cover in general all altitudes between sea level and 1000 ft.).

7 The curve herewith shows the effect of altitude with some 3 per cent difference between the limits noted, as builders can hardly be expected to guarantee that at an uncertain altitude it would generally suffice to state altitude of works, and the committee might insert a table or curve showing the theoretical decrease in engine capacity at different altitudes. This cannot, of course, be applied rigidly to all types of engines, as the effect of altitude differs somewhat, according to the design. In cases of very high altitude, however, as Mexico and the West, ratings would naturally be based on the altitude of the proposed installation. A comparison of records of

the weather bureau shows that the variations in the barometric column due to local atmospheric disturbances are confined to a very definite range. The "power loss" curve drawn in for an altitude of 750 ft. will explain the statements already made in regard to the 3 per cent difference between ratings at East Pittsburg and those made at sea level. A parallel curve drawn through the point of 1000 ft. altitude, shows that a builder rating at this altitude furnishes an engine of 4 or 5 per cent greater capacity when running at sea level than his tidewater competitor.

SPEED

8 Normal speed should be identified as that to be attained at full load, intermediate ratings implying, of course, slightly higher speed, as determined by the speed regulation curve.

GAS SUPPLY

9 The general character of gas and its fuel source should be noted, its effective heat value under standard conditions, 62 deg. fahr. in 30 Hg, dry, the amount of foreign ingredients present in grains per cubic foot, per cent free hydrogen in the mixture, temperature of delivery to the engine and the permissible range in pressure or suction, also the permissible range in heat value or with heat value.

10 This completes the items necessary to an intelligent capacity guarantee.

SPEED REGULATION

11 A definite speed variation should be stated in terms of speed drop from no load to full load. Likewise momentary variations, for a *definite range* of load in terms of normal average speed. Any general expressions for speed regulations are certain to be ambiguous and to be misconstrued. Normal and momentary variation may be segregated as follows: Normal, to refer to change in speed after the governor has settled down to the new load, momentary to refer to the case of violent changes in load before the governor has settled down; in other words, total amplitude of the positive and negative loop of the speed curve. We might distinguish the former as normal rotative speed variation; the latter as instantaneous speed variation, although this would be something of a misnomer. As to the method of measuring speed, if we were sure of a constant speed following a change of load, the most likely method would be to average several minutes by speed counter, but as the speed is liable to change after a certain time, I

should think the average case would best be met by observing with speed counter for a period not less than one minute immediately after the governor has settled down. For instantaneous speeds the indicating tachometer or recording tachograph is the only means available.

12 Parallel operation unless specified in terms of definite cyclical variation (electrical or geometric degrees, + or − mean), can best be covered by a general phrase based upon the supposition that an engine will run in parallel with another of the same characteristics and also with any other prime movers provided such will operate satisfactorily in parallel by themselves.

ENGINE EFFICIENCY

13 I am firmly of the belief that engine efficiency should be expressed in terms of effective heat value, as already outlined in my discussion of Professor Lucke's paper printed in April Proceedings. The reasoning advanced in the National Electric Light Association Gas Engine Report[1] for the adoption of total heat value as the standard, is not to my mind conclusive. Until a combined gas vapor cycle comes into use, will we be confronted with the necessity of using total heat value. For the present at least, let us not confound a definite engine efficiency by reducing the indefinite factor of latent heat of H_2O vapor. Engine efficiencies should be given for full, to half load at least. Other loads are immaterial, as not usually representing commercial conditions of running.

PRODUCER CAPACITY

14 Rating of producer in terms of horse power except in a very approximate way, is a practice that should not be perpetuated. It should be rated upon its ability to gasify coal. To be sure, it would be technically more accurate to rate on British thermal unit output, standard gas. But in practice, this is impracticable owing to the difficulty of measuring the gas and obtaining a good average sample, In actual operation, the size of fuel and relative amounts of ash and volatile, are really the controlling factors, in recognition of which the simplest plan would seem to be to rate the producer on an average coal of known composition, providing a certain margin for increase or decrease in capacity, according to the presence of desirable or undesirable constituents. Should the builder desire to rate on a special

[1] By J. B. Klumpp, Gas Engineer, U. G. I. Co., Philadelphia.

or particular coal, he might for his own protection insert a clause limiting some of the constituents that might vary from time to time from the original sample analyzed.

15 One screen size is not sufficient to secure a specified fuel. We should specify two, a maximum and minimum, to exclude both large lumps and fine dust. As has been brought out before, a mixture of many sizes packs the producer as badly as a very small fuel.

16 The producer should be guaranteed for overload for a period not less than that specified for the engine. As a usual thing, the flexibility of producer operation will more than meet the overload possibilities of the engine.

PRODUCER EFFICIENCY

17 This can only be specified in terms of British thermal unit output involving volumetric measurement in the majority of cases impossible to determine except by calibrating the engine unit and ascertaining the efficiency by heat balance. The time element involved in the complete disintegration of a pound of coal to the form of ash is so great as absolutely to prohibit measurement of gas output, without the use of a meter, except by the power method. Calculations for the Norton test indicate a minimum length of test of 30 hr., during which time between 1 000 000 and 1 500 000 cu. ft. of gas was produced.

18 As we are, therefore, quite dependent upon the engine, as a gas meter, we must necessarily be consistent and determine the efficiency of the producer in the same terms as the engine-effective heat value, i. e., ratio between heat output of standard gas, and heat input of fuel as fired. We have been accused of inconsistency in not using effective heat value of coal as well as of gas. Theoretically, this is correct; practically, it is unimportant, and hardly worth consideration as it involves in all cases the determination of ultimate analyses. Moreover Dulong's formula, I understand, does not apply with any degree of accuracy to wood, peat, lignite and similar combustibles in primitive stages of formation. Until we have a more accurate method of determining effective heat value of solid fuels, I would consider it inadvisable to depart from the old method of bomb calorimeter.

19 In the majority of cases producer efficiency will never be determined, the guarantee being made direct from coal pile to engine shaft. In the case of a complete test, such as at Norton, a heat

balance will generally be obtained, and it then is a simple matter to figure producer efficiency upon either total or effective basis from the calorimeter readings.

20 A method of obtaining producer efficiency in a commercial way has been outlined in my discussion of the Norton test published in Proceedings for Mid-November 1907. From this it will be seen that the efficiency at fractional loads is well sustained and guarantees thereon are hardly necessary.

PRODUCER REGULATION

21 A point not touched upon in Professor Lucke's paper, is the regulating properties both as regards heat value of gas and pressure of the gas delivered to the engine. In at least one type of producer no gas holder is used. In any case the producer builder should specify the range of pressure to be encountered at the scrubber outlet, for a change of load equal to the rated capacity of the engine connected.

22 As to quality regulation, this is entirely a matter of producer design, and is covered by the engine capacity clause "With gas of not less than —— B.t.u. per cubic feet heat value." If a separate clause is needed to cover instantaneous variations with load, this may be inserted, but it is usually unnecessary.

HYDROGEN CONTENT

23 Some phraseology should be used more definite than per cent by volume of gas to indicate the limit placed upon the hydrogen content, this being the principal constitutent of power gas tending to cause trouble if not provided for. The alternative expressions are:

a H per cent volume of gas.
b H per cent by volume of combustible in gas.
c Per cent heat value of gas per mixture.
d H per cent by volume of mixture.

24 The last appears to be the most illuminating. *a* conveys no impression of the commercial value of the gas. Thus coke oven gas with 50 per cent H has about the same per cent H in mixture as anthracite producer gas, with 15 per cent by volume of gas. *b* is better in this respect. *c* presents widely varying values. *d* (based upon theoretical air) shows that those gases with hydrogen content of mixture below 8 per cent or even 10 per cent are suitable for power purposes. I should recommend *d* to be incorporated in all specifications.

PROPERTIES OF GASES

25 As there exists such wide diversity in practice of computing heat values from different temperatures, I would suggest a comparison of authorities and the compilation of a table covering the following:

26 Calorific values of different combustible constituents at 30 in. Hg and 32 and 60 deg. fahr. Air and oxygen required for combustion. Specific heat and weight, this table to cover the ordinary constituents of gases as well as a number of the paraffine and olefine series and other hydrocarbons likely to be encountered, both saturated and unsaturated;[1] the latter form the principal volumetric constituents of the gas at present used at Atlantic Refining Company.

STANDARD GAS

27 I would suggest the adoption of a basis for computing gases either completely saturated with water vapor, or unsaturated (dry). In general practice the volumetric correction for water vapor has been entirely ignored, whereas the correction therefor is appreciable. On the basis of 32 deg. fahr. water vapor is entirely excluded by deducting from the total pressure of the gas the vapor tension of the entrained vapor at the proper temperature, which values may be obtained from any hygrometric table. If we use 62 deg. as a basis, shall we correct for saturated vapor at this temperature to reduce to dry gas, or assume saturated gas? If the latter, this factor should be embodied in a formula or chart for reducing all gases to standard conditions.

AUXILIARIES

28 In view of the widely varying practice in operating engine and producer auxiliaries, a definite statement of the gas, power, heat or coal consumption should be included in the specification. This may be treated separately or collectively with the producer or engine efficiency at the discretion of the builder, but it should always be included in some form.

Note—In further regard to the subject of effective and total heat value, I have figured out four typical analyses for bituminous coals: Pocahontas, Cumberland, Pittsburg and Hocking Valley. The difference between total and effective heat value of the former is 2.75 per cent, and the latter, 3.5 per cent. This difference is so small that it may be neglected. In fact the heats of combustion of

[1] Molecular saturation.

carbon and hydrogen, as applied by various authorities in Dulong's formula, vary almost in this amount. This would seem to discount the criticism of our inconsistency in taking effective heat value of gas and total heat value of coal.

Communication from Mr. J. R. Bibbins

Referring to the probability of the revision of the Standard Code for Testing Gas Engines Mr. Bibbins, in a subsequent communication, suggests that the following points should be considered.

THERMAL STANDARD

29 In outlining a method of obtaining relative efficiency of a steam engine, the code makes use of the term "efficiency ratio," defining it as

$$\frac{2545}{\text{B.t.u. per h.p.}}$$

This is not the efficiency ratio in the generally accepted sense; it is the absolute or kinetic efficiency. The British code includes the same term as thermal efficiency, but elaborates with the term "efficiency ratio," which they define as the percentage heat available in the Rankine-Clausius cycle, which the engine turns into useful work; in other words, efficiency ratio of the Rankine cycle. It seems to me that to apply this term to the fraction above-mentioned, is a misnomer. The term "kinetic efficiency" suggested itself to me some time before Mr. C. V. Kerr presented his paper before The American Society of Mechanical Engineers. Mr. Kerr was criticised for using the term "potential," and I believe, justly, as there seem to be two distinct conceptions in regard to the use of kinetic efficiency. It might be better to refer to the term "absolute," for the fraction above noted, adhering to the term "efficiency ratio" when referring actual consumption to the ideal Rankine cycle.

30 For gas engine work, a standard of comparison is equally as important as in steam, and I believe the right start has been made in adopting the air cycle, in which I am sure you are interested. Why should we not express our efficiencies in terms of efficiency ratio referred to the air cycle, in the same manner as steam efficiency is referred to the Rankine cycle. The same physical conception would thus be observed in both cases. Otherwise we are comparing steam engine performance referred to a certain ideal cycle with gas engine performance referred to the absolute standard.

31 The Rankine cycle assumes that the heat rejected is returned to the boiler at the same temperature as the exhaust steam; i. e., through the agency of a feed heater. Have we a direct analogy in the case of the air cycle, or must we consider the rejected heat as lost?

GAS MEASUREMENT

32 The code does not define the point at which gas should be measured. Assuming the holder as the only practical means of quantitative measurement, pressure and temperature readings must be taken close to the holder outlet, although volumetric correction should be made for the difference, if any. During the Norton test, gas at the holder was from 20 to 30 deg. cooler than at the engine, owing to the fact that the engine supply main ran through the basement for a considerable distance, uncovered, and close to the hot exhaust.

33 Assuming that a standard will be adopted for the hydrometric condition of the gas, either dry at 62 deg., or saturated at 62 deg., will it be necessary to take hydrometric readings of both gas and air entering the engine, along with the other readings, in order to determine the humidity of the mixture before combustion?

34 When analyses and calorimeter tests are being made simultaneously, it is of considerable advantage to take readings with the two sets of instruments at the same time. Otherwise, a considerable variation between computed and observed British thermal units will result, which does not necessarily imply inaccuracy of observation, but simply variations in the quality of the gas between the times of making the two sets of observations.

JACKET WATER

35 Volumetric measurement should be made at inlet to cover loss by vaporization in running the jacket water hot. As the outlet water is composed of several different water circuits running at widely different temperatures, it is necessary that the thermometer be removed far enough from the engine to insure a good mixture, and yet not so far as to make the radiation loss from outlet piping sufficient to lower the reading. It is evident that the outlet thermometer should be so placed as to incur a very low jacket loss.

INDICATORS

36 For gas engine work, would it not be well to recommend or perhaps specify that indicators of the outside-spring type be employed? This will certainly be in the interest of accuracy, as the calibrations will be much less affected by the temperatures encountered. The steam code does not absolutely disbar the taking of successive cards from one indicator after another, and tolerates the use of three-way piping. To be sure, a short indicator connection is advocated. In gas engine work this is all the more important, and there seems to be no alternative but individual indicators and observers. Furthermore, cards should be taken at the same instant by signal, and at least two complete cards should be traced before lifting the pencil. It is seldom that two consecutive cards are exactly the same, as occurs in steam work, and in order to obtain anywhere near consistent results, duplicate and simultaneous cards would seem desirable.

37 Prof. Marks, in his paper before the Detroit meeting, brings up the question, "What is indicated horse power?" which occasioned acrimonious discussion in Germany between Professors Schroeter, Schoettler, Stodola, Riedler and Diesel. It would seem that The American Society of Mechanical Engineers cannot avoid passing upon this intricate subject. Both schools have perfectly logical arguments in their favor. One contends that mechanical efficiency should be only an index of mechanical workmanship, and consequently should include only friction loss, not losses incurred for the purpose of increasing the thermodynamic efficiency. The other school is in practical agreement with the present standard steam code of the United States and Great Britain in deducting all negative cards, giving the gross mechanical efficiency, if we may so term it. There are just two suggestions I wish to make in this connection.

38 First: It is absolutely impossible to discriminate in the two-stroke cycle, between pump loss, considered simply as mechanical friction, and that part which contributed to the higher efficiency of the succeeding power card. The first loss should certainly be included in the term, "mechanical efficiency," the second should be strictly so included.

39 Second: Considering that the four-stroke cycle does its own pump work, whereas the two-stroke cycle requires auxiliary pump work, is it from any conceivable standpoint fair to compare the indicated horse power developed by the two engines by simply taking the

net card in the one case and the gross positive card in the other, or even taking the positive cards in the two cases? Yet, some of the two-stroke cycle engine builders advocate the former under the specious plea that the negative loop of the four-cycle card is traced in connection with the positive card and cannot be disassociated. To make the comparison strictly accurate, can we avoid the use of the net card in all cases, whether pump work is done separately or in the power cylinder?

40 With opinion so evenly divided, the only way out of the muddle is a definite phraseology, and I believe we should adhere to the present practice of deducting all negative areas, but the net area result should not be labeled by the general term "indicated horse power," but by some such term as net, or effective indicated horse power. The difference between the total and effective card in the case of the two-cycle, or the Diesel engine, is entirely too great to be ignored.

SPEED

41 The present code declares "that there is no recognized standard for fluctuating speed," which I presume covers both cyclical variation in reciprocating prime movers and momentary variations due to sudden changes in load. Is there any reason why this ambiguity should not be cleared up?

STARTING AND STOPPING

42 The present code refers to the code on boiler tests for an outline of the method of starting and stopping fuel tests. The latter, however, is practically silent as far as any information is concerned pertaining to the closed ash pit producer; in fact, it is not particularly definite regarding the water-sealed, continuous type producer. The problem is, of course, relatively simple with the small 20 h.p. suction producer holding but a small quantity of fuel. We may start with the producer full of fresh fuel and simply correct for the remaining unburnt carbon at the end of the test; but with a 200 or 300 h.p. suction producer, or one of the Loomis Pettibone type, this method is impracticable, as we found on the Norton test. The only alternative is the flying start and stop, beginning the test after a preliminary run of such length as will insure that all the original coal has passed through to ash. In other words, that a normal fire bed exists with the fuel at different levels in varying stages of combustion. Then a test of any length may be run through with every assurance that the proper amount of coal had been charged, providing the level be

maintained uniform by removing ash and charging fuel regularly. This procedure would require at least a 24 hr. preliminary run, followed by a minimum test run of 24 hr.—preferably 48 hr. If a closed bottom producer will not run continuously this length of time without the necessity of cleaning fires, what procedure is to be adopted? I confess this is a very puzzling question.

43 In testing plants running only ten hours per day, the boiler code suggests that the start and stop be made just before banking for the night. This banking is a very delicate procedure in producer work, inasmuch as standby losses are so small. In fact, they are so small that the judgment of one man as against another would make a very large variation in the apparent standby coal consumption. Assuming that the test covered several standby periods, the problem would not be so difficult, but in the 24 hr. or even 48 hr. test, the difference in judgment of the two men in bringing the fire back to the same condition at banking time, would be likely to vitiate entirely the accuracy of the test for scientific purposes. Some method covering these points in producer testing, should be outlined.

Communication from Mr. H. F. Smith

44 Mr. H. F. Smith submitted the following points to the Standardization Committee, dealing principally with the questions of producer performance, guarantees, and terminology, with the especial intention of eliciting discussion upon these points:

45 Professor Lucke in his excellent presentation of this matter to the Society, makes the following general divisions: For engines: first, capacity; second, efficiency; third, regulation. For producers: first, capacity; second, efficiency.

46 The writer cannot help but feel that while the question of producer regulation may not be of such vital importance as engine regulation it is nevertheless sufficiently important to receive some discussion at this time. We will, therefore, endeavor to discuss briefly, first, producer capacity; second, producer efficiency; third, producer regulation.

47 It is possible to use as a basis for producer capacity

a The horse power that can be developed in a gas engine supplied with gas from the producer.

b The volume of gas generated in unit time.

c The weight of coal gasified in unit time.

d The thermal energy delivered in gas in unit time.

48 To discuss these briefly: *a* This designation of capacity is really only applicable to a complete power installation. The total output in horse power depends quite as much on the engine performance and efficiency as on the producer equipment and this may be dismissed as not applicable to producers but to combined installations of producers and engines under one guarantee.

49 *b* This designation is incomplete as it is meaningless unless qualified by a specification as to the *quality* of each unit volume of gas in which case it is simply a form *d*.

50 *c* This designation is also incomplete unless modified. A boiler furnace will convert into gaseous form large amounts of coal, but it cannot be said to have any capacity as a gas generator in the usually accepted sense. However, this designation in connection with a properly specified *efficiency* could be made a reasonable basis of producer capacity.

51 *d* This method of rating is already adopted as standard for many forms of apparatus. Electrical machinery is perhaps the best known example.

52 The capacity of an electrical generator is designated in kilowatts, this being the measure of the amount of *energy* the machine is capable of delivering for whatever purpose it may be required.

53 It would be the writer's present opinion, therefore, that an expression indicating the thermal energy delivered in gas in unit time could be used as a measure of the capacity of any producer employed for any purpose.

54 A unit for the measurement of producer capacity being assumed, we can begin to consider other matters having influence on the capacity of the plant. Two considerations come at once to prominent notice as exerting a marked influence on plant capacity.

- *a* The net available area for the passage of gas through the apparatus.
- *b* The rate at which the chemical reactions involved in the production of gas can be brought to completion.

55 *a* By this is meant not so much the area of piping and connections as the net area within the fuel bed or the sum of the areas of the interstices between the particles of fuel. This is influenced greatly by the size of the fuel as fired. Large sizes of uniform dimensions give the most porous fuel bed, smaller sizes causing a decrease in available area. The worst possible case is found in a fuel consisting of a mixture of sizes.

56 *b* This appears to depend on, first, superficial area of fuel exposed to the action of the blast; second, time of contact between fuel and blast; third, temperature of fuel and blast at time of contact.

57 The first and second items are intimately concerned with the size of the fuel. In a given machine, increasing the size of the fuel decreases the superficial area of contact and increases the time of contact by decreasing the velocity of blast flow through the fuel bed. Conversely, decreasing the size of fuel increases the area of contact and decreases the time of contact. Evidently, therefore, for a given operating temperature there will be a certain size of fuel that will give maximum capacity, and loss of capacity will result from increasing or decreasing the fuel size.

58 Third item: Obviously, the higher the temperature carried within the producer the more rapidly will the necessary chemical reactions take place and therefore the higher will be the producer capacity. Practical limits in this direction are set by the characteristics of the ash carried in the fuel and also to some extent by the quantity of the ash.

59 The percentage of volatile matter in the coal will also have an appreciable influence on the producer capacity. It might be mentioned here that since we have assumed as the unit of producer capacity thermal energy delivered, it would seem logical to consider all limiting factors in the same light. Accordingly for purposes of comparison it would be convenient to figure the percentage of ash and volatiles not on a pound base but on a British thermal unit basis, e. g.:

	B.t.u. coal	Ash per pound	Ash per 10 000 B.t.u. pound
Pennsylvania	14 000	0.08	0.057
Texas	7 500	0.08	0.107

Likewise, take the volatiles in the above coals

	B.t.u. coal	Volatile per pound	Volatile per 10 000 B.t.u. pound
Pennsylvania	14 000	0.17	0.121
Texas	7 500	0.28	0.37

Thus for a given producer output on the Texas coal it would be necessary to handle twice the ash and three times the volatiles that would be required with the Pennsylvania coal, even though on a

pound base the ash would appear the same and the volatiles in the ratio of about two to one.

60 It is apparent that a definite limit to the practical operating temperatures that may be carried within the producer is set by the fusibility of the ash, i. e., by its tendency to "clinker." This, there fore, becomes a limiting factor as regards producer capacity.

61 It would appear that for the protection of the purchaser, the output of the producer in available thermal energy should be distinctly specified, while for the protection of the manufacturer the limiting conditions of performance will be

a The sizing of the fuel as fired.
b The fusibility of the ash.
c The quantity of the ash per unit of thermal energy in the fuel.
d The quantity of volatile matter per unit of thermal energy in the fuel.

EXAMPLE CAPACITY GUARANTEE

62 This producer shall be capable of delivering in gas 1 000 000 B.t.u. per hour continuously when operating on coal; which shall pass over a screen with $\frac{1}{2}$ in. square openings and through a screen with 1 in. square openings; which shall contain not to exceed 10/100 lb. of ash or 5/100 lb. of volatile matter per 10 000 B.t.u. of thermal value; which will permit an operating temperature of 2000 deg. fahr. without fusing the ash sufficiently to form clinkers.

63 An overload of 25 per cent can be carried continuously with a rise in fuel bed temperature of not to exceed 400 deg. fahr.

PRODUCER EFFICIENCY

64 This involves a consideration of

a Thermal efficiency.
b Efficiency of cleaning and scrubbing apparatus.

65 *a* The thermal efficiency of a producer may be expressed as the quotient of the total thermal energy delivered in gas by the total thermal energy of the fuel used in its production. It is of course essential that the same methods of determining thermal energy be applied to both gas and fuel.

66 *b* The cleaning efficiency of the scrubbing apparatus will usually consist of a determination of the amount of tar, dust or other solid matter carried in the gas.

67 A clause covering the question of efficiency might read as follows: This producer is guaranteed to deliver in gas at rated load, 75 per cent of the heating value of the fuel used. (If efficiency guarantees or other than rated load are required, these should be specified in the same way.)

68 The gas shall be delivered from the scrubbers with not to exceed 0.04 gm. of solid matter per cubic meter.

PRODUCER REGULATION

69 This evidently relates only to the character of the output under varying conditions of load.

70 Very many items seriously affect producer regulation but as nearly all of these are directly under the control of the designer it is unnecessary to discuss them now.

71 The principal irregularities may be classified as

a Variations in gas quality under steady conditions of load.
b Variations in gas quality under variable loads.

72 Specifications covering these points may be worded as follows:

73 When operating at rated load the gas shall have an average heating value of about 135 B.t.u. per cubic foot and shall not vary to exceed 5 per cent either way from the mean value. With a change in load amounting to 50 per cent of the rated capacity of the plant the change in heating value shall not exceed 10 per cent of the mean value.

74 A determination of the quantities used as a basis of the guarantees previously outlined involves primarily a definition of just what shall be considered the thermal content of the gas as produced and of the fuel as fired into the producer. If the producer efficiency is to be accurately determined it is essential that the same standards be adopted in the one case as in the other. There seems to be a disposition among engine builders to favor the use of the so-called low value for gas containing hydrogen, and there is no objection and some possible advantage in this procedure. The thermal energy of the gas may be defined as the heat liberated by the complete combustion of 1 cu. ft. at 760 mm. pressure and 60 deg. fahr. temperature, the products of combustion being cooled to

60 deg. fahr. but water vapor due to combustion of hydrogen not condensed.

75 The definition of the thermal energy of the fuel should be stated in the same way, viz: the heat liberated by the complete combustion of one pound of fuel, the products of combustion being cooled to 60 deg. fahr. but water vapor due to combustion of hydrogen not condensed.

76 In order to be of practical value, a performance guarantee must be stated in terms that are readily determinable. A test to determine producer capacity and efficiency when conducted in the usual and orthodox manner involves the use of quite an elaborate array of instruments and apparatus not to mention the services of a corps of trained observers. In addition if any tolerable degree of accuracy is to be attained the test must extend over a considerable period and should be continuous to avoid the introduction of errors through undeterminable standby losses.

77 Such test involves for

a Producer output:

1 Metering gas { volume readings. pressure readings. temperature readings.

2 Determining calorific value of discharged gas.

b Producer input:

1 Total weight of coal *gasified* during test.

2 Average calorific value of coal gasified.

78 The difficulties in attaining even approximate accuracy in the determination of the first item under each heading are well understood. Determining the volume of gas delivered involves the installation in the gas line of some form of meter. The expense and difficulty in this alone is usually sufficient to prohibit the conduct of an efficiency test. Even when the apparatus is provided, the proper calibration of the instrument and the determination of and corrections of pressures and temperatures involve a great amount of careful and accurate work, and give opportunity for the introduction of serious errors. The greatest difficulty, however, lies in the exact determination of the weight of coal gasified. A producer of 200 h.p. capacity, for example, will contain as a normal running charge about 12 000 lb. of fuel. This would be sufficient to operate the plant at full rated load for six days of ten hours each. In other words, ash from coal fired on Monday morning would not be discharged over the grate until Saturday.

79 The usual method of noting the coal required to cover the shrinkage in volume of the fuel bed produced by a run of ten or twelve hours could hardly be expected to give results of more than approximate accuracy under such circumstances.

80 In order to avoid some of the difficulties involved in determining producer efficiencies in the usual manner, the writer has made use of a method which while open to some objections seems capable of enabling one to secure quickly and easily and without use of any expensive or complicated apparatus at the place of test, data sufficient to permit the determination of producer efficiency with a considerable degree of exactness. It is the writer's wish to present this method briefly for the consideration of the comittee in the hope that if it is not already in general use, it may possibly be evolved and elaborated into a useful and practical method.

81 To consider in the first place the simplest case *versus* the determination of the efficiency of a producer operating on true anthracite coal, i. e., gasifying carbon.

82 For such a determination it will be necessary to secure at the plant:

a An average sample of the coal fired.
b An average sample of ash from the producer ash-pit.
c An average sample of gas generated.

83 *a* An average sample of coal can be easily secured by the usual method of sampling. If great accuracy is desired, each day's supply can be sampled for a week previous to the test, the result being an average sample of the coal then in the producer.

84 *b* A true sample of ash as discharged can be secured by saving all the ash from several days run and sampling by the usual method of crushing and quartering.

85 *c* An average sample of gas can easily be secured by inserting a sampling tube in the gas main and drawing gas from the tube into a container slowly and continuously throughout the time over which it is desired to determine the average efficiency. Simple apparatus for this sampling is so well known as to make further description unnecessary.

86 These samples being secured the observer's work at the plant is completed and the samples can be removed to the laboratory for the completion of the test.

87 Here each sample is submitted to chemical analysis, and if it is not deemed desirable to depend on the analysis for the determina-

tion of the heating value, the coal and gas samples are submitted to test in a suitable calorimeter. To illustrate further, let us assume that the several determinations were as follows:

	Coal by weight		Ash by weight		Gas by volume	Gas by weight	Weight of carbon
H	0.75	C	20	CO_2	8.4	15.9	4.33
C	83.20	Ash	80	O_2	0.3	0.4	
Ash	16.05		—	CO	19.7	23.7	10.21
	——		100	H_2	23.2	2.0	——
	100.00			N_2	48.4	58.0	14.54
					——	——	
					100.0	100.0	

Calorific value, 12 485 Calorific value, 135 B.t.u. cu. ft. L. value standard gas.

	Weight of gas
CO_2	0.010368
O	0.000267
CO	0.015385
H	0.001301
N	0.037955
Wt. cu. ft., gas standard pressure and temperature	0.065276

88 These data being at hand we may reason as follows: The carbon contained in the fuel fired into the producer can escape from the apparatus at two points only: first, through the grate with the ash; second, in the form of gas.

89 We note from the coal analysis that each pound of coal has associated with it 0.1605 lb. of ash. We find from the ash analysis that for every 0.1605 lb. of ash discharged into the ash-pit, there is carried along with it $0.20 \times \frac{1605}{8000} = 0.0401$ lb. of carbon. Evidently, therefore, of the 0.832 lb. of carbon contained in each pound of coal 0.0401 lb. are wasted in ash and 0.7919 lb. converted into gas. We also note that 1 cu. ft. of this gas when reduced to standard conditions of pressure and temperature weighs 0.065276 lb. From the gas analysis we find that 14.54 per cent by weight is carbon. There is, therefore, $0.065276 \times 0.1454 = 0.00949$ lb. of carbon in each cubic foot of gas produced.

90 Since 0.7919 lb. of carbon are gasified for each pound of coal fired, we conclude that there are generated $0.7919 \div 0.00949 = 83.4$ cu. ft. of gas per pound of coal. Since each pound of coal represents 12 485 B.t.u. we would fire into the producer $12\ 485 \div 83.4 = 148.7$

B.t.u. in coal for each cubic foot of standard gas produced. The heating value of the gas being 135 B.t.u. per cubic foot the producer efficiency would be $\frac{135}{149.7} = 90$ per cent.

91 In extending the application of this method to producers gasifying bituminous fuels the process is complicated by the necessity for establishing a definite ratio between the fuel burned and the condensible hydrocarbons, lampblack, etc., that pass away from the apparatus in the scrubber water or remain deposited in various cleaning devices. It is not a matter of very great difficulty, however, to determine this with a very fair degree of accuracy, and any desired degree of accuracy can be attained by increased care. A convenient method consists in drawing off a definite volume of hot unscrubbed gas and determining carefully the nature and weight of condensable matter and particularly the weight of deposited carbon per cubic foot of gas. This weight of carbon added to the weight of truly gaseous carbon, gives the true weight to divide into the carbon content of the coal in determining the volume of gas generated per pound of fuel fired. The remainder of the calculation remains as above illustrated.

92 It will be noted that, unless the heating value of the coal is to be determined from analysis, it is not necessary to carry the analysis to completion, a determination of carbon and ash being all that is required.

93 This discussion of ways and means is perhaps apart from the real purpose of the committee, but its apparent simplicity as compared with the unwieldly heat balance determination is my excuse for calling it to your attention. It is perhaps open to more serious criticism than has yet come to the writer's attention.

94 Very simple and satisfactory methods are already available for determining the purity of producer gas and as well for detecting and measuring variations in gas quality.

Communication from Mr. J. B. Klumpp

95 Mr. J. B. Klumpp, a member of the Gas Engine Committee of the National Electric Light Association, submitted the following communication upon the subject of the heating value of gases.

96 The heating value of gases has generally been expressed as the gross or total heating value, although the expression "net heating value" has been at times erroneously used in this connection. The true heating value of a gas is undoubtedly the total heat you can get

by burning this gas and reducing the products of combustion to the initial temperature of the inflowing gas and air; in other words, the total heat produced by the gas burning when the water vapors formed in the products of combustion are condensed to a liquid. This is the value obtained when burning the gas in a calorimeter that is operated according to certain definite specifications.

97 In many experiments with gas calorimeters I have found that the surest way to get confirming readings is to have the gas, water and air all at the same temperature, preferably about 60 deg. fahr., and to operate the instrument so that the products of combustion leave the calorimeter at the same temperature as the inflowing gas and air. Owing to the construction of the best forms of calorimeter in use today, it is sometimes necessary to lower the temperature of the incoming water slightly to obtain these results, but this should not materially affect the readings, providing we maintain the gas, air and products of combustion at the same temperature. Maintaining these temperatures alike reduces the likelihood of error in heating excess quantities of air, or the water vapors entrained in the air.

98 Many of the gas engine manufacturers suggest that the latent heat in the products of combustion is never available for power in a gas engine; this is undoubtedly true, but so is much sensible heat, as it is impossible to discharge the exhaust below certain definite temperatures.

99 The so-called "net" or "available," or "effective heating value," as used in writing contracts or guarantees, are misnomers, and economizing appliances could be attached to some appliances to produce results over 100 per cent, if these terms were considered. If it is necessary to use such a term, why not consider the expression "effective power value" as expressing this lower value, but not the net heating value?

100 The latent heat in the products of combustion is not lost until the gas and air necessary for combustion are raised at least to 212 deg. fahr. when the sensible heat then amounts to a considerable proposition of the total heat lost, and should the exhaust of the calorimeter, water heater, gas engine or any appliance reach a temperature as high as 500 deg. fahr., it will be found in many commercial gases that the sensible heat lost will more than exceed the latent heat lost; then the net heating value would be very much lower than that generally quoted as the net value.

101 It might be well to express this "effective power value" as being the true heating value, less the latent heat contained in the water vapors in the products of combustion.

102 I think your committee will agree with me that the present net or effective heating value is a wrong term, and that, as there is no question among the interested parties as regards the scientific description of this term, we might get together on a suitable expression that will describe the heating and power values in a true manner.

Communication from Mr. Arthur West

103 All of the suggestions made by Mr. Bibbins had been previously discussed during our mutual connection with the Westinghouse Machine Company. They represent very closely my own opinions upon the matters in question. I do not think I could express my views along this line any better than Mr. Bibbins has done and would therefore ask you to consider them expressions of opinion in a general way from me as well as from Mr. Bibbins.

104 Referring also to the copy of letter sent to you by Mr. J. B. Klumpp, I have the following comment to make:

105 The term ordinarily in use in gas engine and producer contracts is gas having so and so many effective British thermal units. It seems to me that Mr. Klumpp's objection would be met if all parties interested agreed that the term "effective British thermal units" should be construed to mean British thermal units effective in producing *power*. The difference of opinion existing on the subject seems to me to originate in the fact that some users of the term mean British thermal units that are effective in producing heat and others British thermal units effective in producing power. If a rigid definition of the term "effective British thermal units" were thus agreed upon, no confusion could result.

106 The heat in the exhaust from a gas engine is not now available for power, hence I desire to support the contention of the gas engine builder that this reckoning be based only upon British thermal units that are effective for power purposes. It is of no interest or value to the gas engine builder or user to know that the exhaust contains heat in the form of uncondensed steam produced by chemical action in the gas cylinder, if such heat cannot be converted into power.

107 On the other hand, there are parties who employ gas for heating purposes exclusively, who properly use guarantees based upon the total operating heat value of the gases, since all the units are available for producing heat but only a portion of them for producing power.

DISCUSSION AT GAS POWER SESSION

SEMI-ANNUAL MEETING, DETROIT

THE BY-PRODUCT COKE OVEN

By W. H. Blauvelt, Published in March Proceedings

Mr. J. R. Bibbins It is fortunate that this subject of by-product coke ovens has been brought before the Society by Mr. Blauvelt at this time, especially with the promise of a demonstration upon so large a scale. The by-product coke oven offers so many interesting phases of commercial development that it is difficult for the layman to understand the reasons for its somewhat limited present application. Yet a more careful study of the operation of the large by-product coke plant soon reveals the necessity for the coördination of many important factors, any one of which might operate to disturb the balance of the whole.

2 To the outsider the most interesting phase of coke manufacture is the production of by-products. The Detroit installation serves as an illustration of the use of by-product gas for illuminating or power purposes. It is the popular impression that the sale of by-product gas for this purpose represents clear profit without any difficulties or embarrassments encountered in the guarantee of an uninterrupted supply of gas. Although there are many factors to be considered it is true that under certain conditions power generation from by-product coke gas is commercially profitable, both to the supply company and to the customer. At a reasonable price for gas, power may be generated at a cost per kilowatt so far below the cost of steam power as to preclude the use of steam entirely.

3 In this connection some data regarding the first coke oven gas installation of large size in this country may be of interest. This development has taken place at Lebanon, Pa., where two large by-product coke plants are in operation, one of the Semet-Solvay and the other of the United Otto type. Until about a year ago there was no opportunity to dispose of the by-product gas from either of these

plants, and all the surplus was allowed to go to waste. In 1907, two power projects were developed—one to operate the works of the American Iron and Steel Company, and the other to establish a large gas engine plant near the ovens to furnish power for the ovens and mines supplying them, and to transmit a large amount of surplus power to the city of Lebanon for public utilities. The first of these plants was installed in 1907 and has been in continuous operation since, from gas piped several miles under moderate pressure from the Semet-Solvay coke plant. The second project will be actively pushed forward within the next few months. Both employ horizontal double-acting engines of the twin-tandem type of 500 and 1200 h.p. capacity respectively.

4 At the time of this first Lebanon installation very little was definitely known concerning the possible success or failure of this kind of gas, as the only precedents in this country were a small vertical single-acting engine at Syracuse and a larger engine at Camden, N. J., which was in commission for several years, operating on poor gas from the last half of the coking run.

5 The results of a year's operation at Lebanon have been extremely satisfactory. Trouble anticipated from excessive hydrogen content has not developed, and after some experimenting for sulphur removal, no serious trouble has resulted from the somewhat high sulphur content. At the outset, the builders stipulated a maximum content in the gas of 50 per cent hydrogen by volume and 0.02 gr. of sulphur per cubic foot. Both these limits have been exceeded without resulting trouble. Although pooled gas (i.e., run-of-oven) is supplied, the hydrogen content at times runs as high as 66 per cent by volume, and the sulphur 1½ gr. per cubic foot.

6 This sulphur exists in the form of H_2S and CS_2 in varying proportions. The gas is purified in rectangular oxide purifiers containing iron oxide sponge prepared on the ground. By giving proper attention to this apparatus and renewing one layer of oxide each week, the sulphur may easily be reduced to 80 or 85 per cent of the original content, which runs from 2½ to 4 gr. per cubic foot in the crude gas. This leaves about 0.6 gr. sulphur in the gas supply to the engine, much more than was originally considered safe.

7 As to the effect of sulphur upon the engine, it apparently passes through the combustion cycle without difficulty, provided three simple remedies are employed:

a To insure absolute separation of the water and gas around the cylinder valves or packing;

b To flush the rods thoroughly with continuous streams of engine oil;

c To maintain the piston cooling water circuits at a fairly high temperature, sufficient to prevent condensation and the consequent formation of acid.

8 In all modern designs, a continuous return lubricating system is employed. It is a simple matter to provide a small stream of oil in front of each packing cage, which returns to the filter and is again pumped through the system without loss.

9 A fairly high piston water temperature is advantageous, not only from the sulphur standpoint but to facilitate lubrication as well. In fact most of the packing trouble experienced in the early days of the horizontal design was found to be due to running the rods too cold, as a matter of precaution.

10 During the early experience of the Lebanon plant, the sulphur reduction decreased for some time to a point as low as 50 per cent, or 2.0 gr. per cubic foot at the end of the week. Recently a measurement of the rod diameters showed that the reduction during the year had been extremely small, and at one packing could hardly be detected. This seems to indicate that in a proper design and with necessary precautions the sulphur problem is not so serious as anticipated. Recently this engine was operated six hours on crude unpurified gas containing over 3 gr. of sulphur, and has been so operated at intervals during the past year. With normal lubrication the rods tended to run dry in places, which might cause considerable trouble at the packing, but by flushing the rods with a small stream of oil, this trouble was immediately corrected.

11 One important phase of the Lebanon situation is the relative value to the coke oven operator of principal and by-products, and herein lies the crux of the application of coke oven gas for power purposes. The power user of course desires a guarantee of uniformity of quality and continuous supply; and this necessitates the operation of the ovens for the purpose of delivering the by-product even in periods of decreased demand for coke, tying up the capital of the operator in stored coke, to his serious embarrassment. At Lebanon the gas is supplied by contract that fixes the rate, but not the supply, which is dependent on the operation of the coke oven plant. This provides cheap power and heat for the consumer, and relieves the coke oven operator of the above objection. On the other hand, this arrangement would be entirely unsatisfactory to a consumer contracting for a definite service unless he took the precaution of installing

a relay producer equipment. Such a plant has already been installed at Lebanon to provide for a contingency of this nature, and at the prevailing low price of producer apparatus would certainly be cheaper in the long run than to employ steam power, providing the coke oven gas could be obtained at a reasonably low figure in comparison with coal. This whole question therefore resolves itself into an economic one which both producer and consumer must solve.

12 Another point of interest in regard to the Lebanon installation is that it is an alternating current system with generators solid coupled, and operating at a frequency of 40 cycles per second. The unit operates regularly in parallel with an adjoining steam plant containing both tandem and cross-compound steam engines. At this intermediate frequency no trouble whatever has appeared in parallel operation.

Mr. C. M. Barber Mr. Blauvelt's paper is certainly a very valuable contribution to our knowledge of coke manufacture. The writer would like, however, to advance a few ideas in regard to the *cooling of coke.*

2 That the oven heat of coke from by-product ovens can be removed without the use of water is a proposition that the writer has demonstrated to his own satisfaction. The proof is based on the fact that the largest single pieces that are usually discharged from the Semet-Solvay oven, if isolated from contact with the mass, will cool without combustion in the open air.

3 A charge of good coke when pushed from the oven while it has a high temperature is not undergoing combustion. The term "quenching" implies combustion. We would substitute the word "cooling" on the ground that the coke when ready for discharge from the oven has parted with all those components of the coal which would ignite when exposed to the air. The coke itself, if properly handled, does not ignite. The process therefore it seems to us is one of cooling rather than of quenching.

4 If the mass is simply spread out so that the largest pieces practically lie by themselves the coke will give up its heat just as a newly rolled bar of steel cools in the open air.

5 By discharging the coke from one oven directly upon a conveyor, say, 6 ft. wide by 150 ft. long, the average thickness would be less than 3 in. Considering a block of 30 ovens and allowing 22 hours for coking, an oven is discharged on an average every 45 minutes. Five such conveyors could be arranged to give a cooling period of

about three hours which would simply mean that the coke is detained about three hours on its trip from the ovens to the furnace.

6 In Fig. 1, *A*, *B*, *C*, *D* and *E* are conveyors, each about 150 ft. long by 6 ft. wide. They are adapted to receive the coke from a block of 30 Semet-Solvay ovens. *A* and *B* displace the ordinary coke car and its track. The top of *A* is about one foot below the floor of the oven and receives the charge of any oven of the block. *C*, *D* and *E* may be located directly over *A* and *B*. *G* is an elevator having buckets adapted to receive the coke from *B* and discharge it on to *C*. *A* and *B* are coupled to work together at the same speed which is variable and controlled.

7 The operation is as follows: The contents of an oven is slowly discharged by the coke pusher in the usual way onto the conveyor *A*, at any point *X*. The man operating the conveyors regulates the

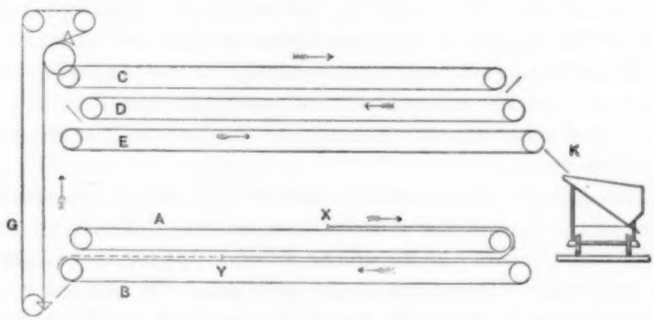


FIG 1 ARRANGEMENT OF CONVEYORS FOR RECEIVING AND DISCHARGING COKE

speed of *A* and *B* so as to distribute the entire contents of the oven over about 150 ft. of conveyors *A* and *B*.

8 This carries the charge to about the point marked *Y* on conveyor *B*. He then continues the movement at any speed he wishes until the point *Y* has arrived at the discharge end of conveyor *B*, when he locks his lever at a speed just sufficient to discharge the entire contents of the one oven from conveyor *B* on elevator *G* in about 45 minutes, which will leave conveyors *A* and *B* empty when it is time to use them for the next oven.

9 The operation already described will require from about three to five minutes, or the usual time of discharging an oven plus the time for completely covering conveyor *B*, making the whole operation not more than an average of about six minutes.

10 Elevator *G* and conveyors *C*, *D* and *E* are all run at one uniform speed of about 150 ft. in 45 minutes, and the coke is finally discharged at *K* into a car or in some cases directly into the bins at the blast furnace.

11 The above is a general description, showing the lines upon which the apparatus can be worked out.

12 The advantages of cooling coke from the oven temperatures to that required for handling without the use of water are almost too apparent to require noting here. We may, however, call attention to the effects of dashing upon the incandescent coke quantities of water sufficient to cool it to blackness in a few seconds. Coke has a cellular structure. The cell walls are hard and vitreous. Struck with a hammer good coke will often ring like crockery. It contracts considerably on cooling. The almost instantaneous cooling cracks and breaks it very much as the same treatment would break almost any like substance. We know that glass, furnace slag, cast iron and many other substances fly to pieces when suddenly cooled with water. The watering of incandescent coke is always accompanied by cracking, snapping sounds caused by the disintegration of the coke. We know too that a considerable amount of the breeze is caused by the effect of the water.

13 The delivery of coke to the furnace bin with a low content of moisture is the aim of the coke maker, but the difficulty of getting the water to the interior of the pile as it lies on the car is considerable. In fact no water reaches the inner part until the outside is overwatered and the ideal conditions of leaving just sufficient heat to completely dry the coke before the car reaches the bins is practically impossible to realize.

14 We believe that it will be conceded that coke cooled without water is harder, stronger and will stand dropping into bins with less breakage. The quantity of breeze will be less, and since the coke is in larger pieces and not full of shrinkage cracks, there will be less of it lost by the dissolving action of the carbon dioxide gas in the upper zones of the furnace.

15 The increased value of the coke will far more than doubly repay the probable cost of 750 ft. of conveyors moving at a speed of 3 ft. per minute.

Mr. C. G. Atwater The discussion which I have to offer on Mr. Blauvelt's complete and well considered summary of the by-product coke oven situation up to date, consists of the operating figures of

the industry in this country for the year 1907, which naturally were not available at the time the paper was written. These figures have been compiled from reports made to me by all or nearly all, of the oven operators. The courtesy of these gentlemen, among whom is Mr. Blauvelt himself, it is proper for me to acknowledge here.

2 The total figures are given in the following table:

OPERATING DATA FOR BY-PRODUCT COKE OVENS IN U. S. FOR THE YEAR 1907

Coal carbonized, net tons	7 391 285
Coke produced, net tons	5 492 425
Tar produced, U. S. gallons	53 393 495
Ammonium sulphate, or sulphate equivalent, produced, net tons	62 700
Gas produced and sold, cu. ft. (from reports and estimates)	12 588 000 000
Total gas produced beyond that used for oven heating, cubic feet estimated	15 500 000 000

3 These items I will discuss in order: The coal carbonized in 1906 in by-product ovens, as given by the U. S. Geological Survey, was 6 192 068 net tons, or something over a million tons less than was carbonized in 1907. Similarly, coke they produced in 1906 was 4 558 127 net tons, very close to a million tons of coke behind the 1907 figures. The yield of coke per ton of coal, which was 73.6 per cent in 1906 was 74.3 per cent in 1907, this difference being probably due to the increased carbonization of low volatile coal yielding a high coke percentage, in the last year.

4 The total coke production of the United States was but three or four hundred thousand tons greater in 1907 than in 1906, as nearly as preliminary estimates can ascertain; therefore it may be seen from the previous statement that the gain in total coke production was all due to the by-product oven, and that the production from beehive ovens actually fell off to some extent. Up to the last two months of the year the coke production from both sources promised to be ahead of any previous year, but the industrial depression intervened, and the beehive ovens, being better adapted to shutting down and starting up again on short notice, naturally suffered first. If we assume that the total production was 37 000 000 net tons, which is probably beyond the actual figure, the proportion of by-product coke produced is very close to 15 per cent. For the year 1906, as shown by the figures given in Mr. Blauvelt's paper, the percentage was 12.5 per cent, so that the year 1907 will probably show a good advance here, as was to be expected from the other figures. If we review the figures for years past, as given in the paper above-mentioned, we find that such an advance from year to year has

occurred several times, and that the by-product coke oven has made steady progress since its introduction.

5 The production of tar in by-product ovens in 1905 is given as 36 379 854 gal., the report on this product, as well as that on gas and ammonia, having been omitted for the year 1906 by the Geological Survey. The figure given in the table for 1907 shows an increase of something over 17 000 000 gals., or about 47 per cent for the two years. The actual yield of tar per net ton of coal carbonized was 7.22 gal. in 1907, 7.86 gal. in 1905 and 7.77 gal. in 1904. This decrease in tar per ton of coal is due to the increasing use of the lower volatile coals, to which cause the increased yield of coke was also ascribed. Though these coals give lesser yields of gas and ammonia, as well as tar, they give an excellent quality of coke as well as a larger quantity, so that their use will probably increase as time goes on.

6 The production of ammonia, in the form of ammonium sulphate or its equivalent, amounted to a little less than 62 700 net tons, or about 17 lb. of sulphate per net ton of coal carbonized. When it is considered that this figure covers all the vicissitudes of operation throughout the whole industry, it must be regarded as a very creditable one indeed, though doubtless still capable of improvement.

7 The figure for gas produced and sold cannot be regarded as strictly accurate, as some of the items included in it are estimated. It is intended to represent the amount of gas from by-product ovens that actually came upon the market, on the same basis as city gas or natural gas. Some of it was used for heating steel furnaces or glass furnaces or for steam making, or for direct power generation in the gas engine, but a good proportion of it was used in general city distribution, either alone or in combination with carburetted water gas. It does not include gas from all the plants, however, as some are operated on low volatile coal which produces only enough gas to heat the ovens themselves properly, and at other plants the gas is not sold at all, but used for raising steam or for other purposes around the plant, or is wasted. The amount of coal carbonized at the plants reporting their gas as all or partly sold was about 5 600 000 net tons, or there was about 2250 cu. ft. of gas disposed of per net ton of coal carbonized. It is believed that this figure is, if anything, below the actual facts, rather than above.

8 The figure for gas total produced beyond that needed in heating the ovens is intended to include the gas used for steam raising or other purposes around the coke plants themselves, wasted for lack

of an immediate use or by leakage in distribution, or that for any other reason cannot be included in the figure for gas sold. The amount is estimated in many instances, but such estimates are made on a very conservative basis.

9 To turn to another phase of the by-product coke oven subject, viz., the use of coke oven gas in the gas engine, I would like to ask a question. It has been stated to me as a point developed in experience with a coke oven gas containing considerable sulphur that the piston rod of the engine lost its lubrication and became dry when the cooling water in the rod was allowed to get too cold, but that if the water was kept up to a moderately high temperature, the oil adhered better and the danger of scoring the rod appeared to be less. Such an action might be attributed to more condensation of acidulated moisture on the cooler rod, hence more removal of the oil than when the moisture and acid vapors remained volatile and went off in the exhaust. I would like to know whether anyone has had experience with sulphurous gases that bears upon this point, and whether the explanation suggested is in accordance with our knowledge of the somewhat obscure reactions that take place in the cylinder of an internal combustion engine.

Prof. R. H. Fernald Referring to the action of sulphur on gas engines, we operated the engines in the government testing station in St. Louis for 2½ years and used no purifier. We used bituminous coals, lignites and peats. The highest percentage of sulphur in any fuel used was 8.2, contained in California lignites. These fuels were used without cleansing as far as the sulphur was concerned and we had no difficulty at all with the engines so long as we carried out the scheme of Mr. Bibbins for keeping the water away from the sulphur. We carried the gas directly into the engine and discarded the purifier. When the plant was removed we left the purifier in St. Louis and have made no use of it since.

Mr. John C. Parker Two points brought out by Mr. Bibbins' very interesting discussion and his citation of the Lebanon plant present themselves to the speaker as being of especial interest.

2 Mr. Bibbins has referred to the sale for power purposes of such by-product gas as may occur and to the use of this by the customer as it may be generated. I should like to inquire what source of supply is used during the period when the coke ovens have not a sufficient supply of by-product gas to furnish the necessary demand of the gas

engines. It is economically possible for the customers to carry stand-by producer plants?

3 The other point involves the determination of price in the sale of any economic utility. Two things determine the sale price: the cost of production and the value of the utility rendered. In the case of by-product sale the cost of production is indeterminate between that of the fundamental product and the by-product, but the compound cost of production is very definite, depending upon the ratio upon which the fundamental product and by-product may be sold. If there is a large demand for one of the products, which, for the sake of discussion, we will call the fundamental, the price of this will tend to rise and in order to dispose of the by-product the price of it must be marked by the value—in competition—of the utility rendered. If in the early days of the business this price is placed to meet competition it is likely to be set at a very low figure which may induce a comparatively large demand. It would be difficult, however, to raise the price as in the sale of any public utility there is a general prejudice against the raising of price for any reason whatever. Where by-product gas is sold in competition with other forms of power it might seem that there would be a danger of over sale resulting in forcing the main product to become a by-product as has happened in so many industries. In the case of the sale of a public utility, such as gas for power purposes, is it possible to refuse arbitrarily the sale of power gas before such a condition is reached; or is a different method used to prevent the growth of a demand which may necessitate the manufacture of the fundamental product in quantities beyond the economic market value, thus running down the profit on the fundamental product?

Mr. G. J. Rathbun A manufacturer of lubricating oil stated to me that he was having trouble with lubrication of piston rods on a gas engine, where the gas was known to be high in sulphur. I advised him to run hot water through the rod instead of cold and he afterwards stated that this overcame the difficulty.

POWER PLANT OPERATION ON PRODUCER GAS

By Godfrey M. S. Tait, Published in June Proceedings

Mr. C. W. Lummis I have seen several of these plants operating and they work very well. The company with which I am connected has built a few producer plants which have been equipped with this

apparatus and they are giving very satisfactory service. On account of the large amount of nitrogen which is introduced with the engine exhaust into the producer the efficiency of the producer is somewhat lower than when the heat of the primary combustion is used to decompose steam. Improved methods of regulating the steam supply to the producer have resulted in such fine running plants that it would be a mistake to assume it is necessary to install the system advocated by Mr. Tait in order to have an entirely successful plant.

2 The idea of putting the gas under pressure is a very good one but the simple fan exhauster is in my opinion preferable to the positive exhauster on account of increased ease of control. This is especially desirable when more than one engine is to draw gas from one gas line.

Mr. H. F. Smith There is no question but that a variation in the percentage of hydrogen in the gas is the cause of serious difficulty in operating gas engines. Our company solves the problem, not by eliminating the hydrogen, but by controlling it. I am inclined to believe that the use of exhaust gases containing a large amount of inert nitrogen would result in a material loss in producer efficiency.

Mr. E. P. Coleman I do not know that such experience as I have in mind is exactly pertinent to the subject of gas power, but it has to do with gas producer practice for an open hearth furnace in which an attempt was made to substitute carbonic acid for steam as the endothermic reagent.

2 The experiment, at least during the time of my connection with it, was not successful, and I am credibly informed that it was finally abandoned. The gas seemed to be of too low a heating value for that kind of work, and we were unable to continue the operation of the producer for a length of time sufficient to get any quantitative results from it. That is to say, we were able to continue only until the furnace became so cold that the melter refused to operate longer and a return would be made to normal operation with steam.

3 Other troubles were due to the fact that in order to equip for the experiment cheaply, no provision was made for cooling the gases taken from the furnace stack, and in an attempt to force a sufficient quantity of the hot mixture through the usual air-supply piping, the fan was finally speeded up to a peripheral velocity of about three miles per minute. Then again during the operation of the reversing valves, some gas would escape to the stack-flue, thus producing an explosive mixture which would occasionally get into the fan piping.

MR. C. J. DAVIDSON I understood Mr. Tait to say that he experienced difficulty in obtaining results which have been obtained in Europe because of the fusibility of our ash. If he refers to bituminous coal I would like to ask Mr. Tait where and to what extent it has been found successful in Europe, but if it applies to anthracite coal I withdraw the question. I spent some time there about a year ago and endeavored to get some information upon that subject, but I was unable to find anyone who was even enthusiastic over the possibility that bituminous fuel would ever be a satisfactory fuel for gas engines.

MR. H. W. PECK I have known several cases where the producer gas engine plant has not proven its reliability. An important point in this connection is the length of time plant will operate without shut downs. The greatest drawback apparent in any published reports on gas engine plants is their unreliability. I would like to ask Mr. Tait the remedy for this unreliability in producer gas engines, and whether the performance is becoming such as to disprove these reports.

2 At the National Electric Light Association convention at Chicago the amount and character of attendance and supervision desirable for a gas engine plant was made very prominent. It seems to be a general opinion that a good machinist or a good all around mechanic secures much better results than a steam engineer in operating such a plant. He is better equipped mentally and better adapted to meet the troubles which he is liable to experience.

MR. F. H. STILLMAN A very good gas engineer has told me if he wanted a good man for a gas engine plant he would not take one of the steam engineers as they almost always made a failure. He wanted me in my own case to take a man of intelligent character but not an engineer, and he would guarantee that within a week he would be a much better overseer of the plant than would an engineer. In reference to Mr. Tait's producer, I will say, that at the time of the meeting in New York I was just starting up a producer of a peculiar type. The producer has not been a success for the reason that the interior is practically one piece of fire brick and was intended to use fine coal. Clinker formed, and when we did break through the clinker and got the coal in the producer running as it should we burned holes through 12 in. of the fire brick bed. Mr. Tait is now at work on it

and we shall probably have the producer operating within the next month under his system.

Mr. H. W. Jones I think nothing has been said about illuminating gas engines, and I know of no branch of the business so badly in need of conference as this one. In Chicago there are about 700 gas engines, 550 operating with illuminating gas and 150 with natural gas. Why do some of these engines give so little trouble and others so much, and what can we do to change this condition? I have oversight of all these engines and we have yet to find one complaint either as to reliability or cost of gas where the engine has had proper care and attention by the maker as well as the user. When complaints are made, we find that low compression or faulty ignition or some mechanical defects are responsible.

2 What can we do about it? I have reason to expect that some members of this Society will interest themselves in the matter, for we need them in illuminating gas engine work.

3 In reference to Mr. Tait's producer gas plant, I desire to say that I have visited the plant and the engine was operating perfectly and, it seemed to me, the producer as well. During my stay in the engine room (about three hours) the engineer was not called up from his chair on account of engine or producer.

Mr. J. R. Bibbins Mr. Tait appears to consider of great importance the suppression of hydrogen in producer gas in order to secure an absolutely fixed ignition point, and even considers this more important than the production of a gas of high heat value. I believe he has over-estimated the importance of securing a fixed ignition point, and even if this were desirable, it is accomplished at too great a cost. Moreover the elimination of hydrogen would be a very small factor in effecting such regulation in gas quality as to make it possible to run with a fixed ignition. Producer gas unavoidably varies in composition and heat value and the engine operator compensates for this variation by changing the mixture and ignition. So many other factors bear upon the point of ignition that the elimination of hydrogen is but a small step in that direction.

2 But most important, Mr. Tait raises a question upon which he should further enlighten us. As I understand him, it is plain that in the combustion of a complex mixture of gases, each having a different rate of flame propagation (that is, ignition point), the more rapid gases burn out first and the more sluggish last; in other words the mixture

is progressively "honeycombed" as the combustion progresses, necessitating, as is claimed, a different time of ignition for each constituent. It seems hardly possible that with finite speeds of flame propagation and the intimate mixtures of various combustible constituents, any other than a resultant effect would be produced; i.e., an average speed of flame propagation dependent upon the relative proportions would obtain, and with it a single definite ignition point would be required.

3 This is evidently a precise scientific problem, and if it has been proven that combustion is not synthetic, so to speak, the results would be of extreme interest. But the gas engine in actual operation seems to recognize only average combustion effect and the adjustment of the ignition point is a relatively simple matter.

HORSE POWER, FRICTION LOSSES AND EFFICIENCIES OF GAS AND OIL ENGINES

BY PROF. LIONEL S. MARKS, PUBLISHED IN MAY PROCEEDINGS

MR. HENRY HARRISON SUPLEE As a discussion upon the paper of Prof. Lionel S. Marks, the Secretary of the Gas-Power Section of the Society was requested to contribute to the published discussion an abstract of the controversy which took place before the *Verein deutscher Ingenieure*, concerning the true mechanical efficiency of the modern internal combustion engine.

2 As that discussion covered many pages of the *Zeitschrift* of that Society, and aroused a somewhat acrimonious controversy, it has been thought best to give simply a summary of the essential points involved as forming an introduction to the paper of Professor Marks, and to refer those who may be interested in the details of the discussion to the reports of the German Society.

3 At the forty-fifth annual convention of the *Verein deutscher Ingenieure*, held at Frankfort-am-Main, on June 6, 1904, Professor Riedler delivered an address upon the subject of large gas engines, in the course of which he called attention to the fact that the determination of the mechanical efficiency of such machines presented important differences from the accepted conditions for the steam engine.

4 In the case of the steam engine, the mechanical efficiency is taken as the ratio between the brake horse power and the indicated horse power. For the earlier four-cycle gas engines this practice

had been followed without comment. When, however, the two-cycle gas engine came into use, the charge of air and gas was partially compressed in a separate cylinder, and the question arose as to whether the work absorbed by this compression cylinder should be deducted from the indicated power of the engine, or whether it should be left as a separate matter. Professor Riedler cited a report made by Professor Meyer, of Berlin, upon a 500 h.p. Oechelhauser engine, in which the pump horse power had been subtracted from the indicated horse power before the computation of the mechanical efficiency.

5 That this question possessed more than an academic significance appears in the fact that had the resistance of the compression pumps not been deducted from the indicated power before the computation of the mechanical efficiency, the latter would have been nearly 8 per cent lower than appeared in the report of Professor Meyer.

6 The discussion upon the paper of Professor Riedler led to the appointment of a special committee, consisting of Messrs. Schöttler, Stodola and Schröter, to determine the correct procedure in such cases. This committee reported in favor of the procedure of Professor Meyer, while at the same time rather evading the question as to the scientific basis for his method.

7 Professor Riedler returned to the charge, and maintained that the indicated power is the actual power developed in the cylinder and that the work absorbed by the pumps constitutes a part of the resistance of the engine.

8 Professor Stodola, whose views are certainly entitled to full consideration, maintained that Professor Meyer's practice conformed to that obtaining with the steam engine, at least so far as the high-pressure engine is concerned. With the condensing steam engine there is a difference in practice: if the air pump is driven by the engine its resistance is not deducted from the indicated cylinder power; if the air pump is independently driven, its resistance is subtracted. In general, Professor Stodola was inclined to believe that, for two-cycle gas engines, the pump resistance should be deducted; the mechanical resistance of the engine being assumed to include only frictional resistances, so far as the computation of the mechanical efficiency is concerned.

9 Mr. Diesel suggested the subdivision of the various quantities, thus: indicated power = the full power represented by the indicator diagram of the working cylinder; useful power = the power delivered at the engine shaft; effective power = the sum of the useful power and

the pump resistance; mechanical efficiency = useful power divided by indicated power (thus supporting Professor Meyer); dynamic efficiency = effective power divided by indicated power; pump factor = pump resistance divided by the indicated power.

10 The whole subject settled itself into a matter of definition, as to whether the compression pumps formed an integral part of the engine or whether they should legitimately be considered as auxiliaries.

11 For a full report of the matter reference should be made to the *Zeitschrift des Vereins deutscher Ingenieure* for 1905, as follows:

a Original paper of Professor Riedler; Report of Professor Meyer to Messrs. A. Borsig upon a 500 h.p. Oechelhauser gas engine; Report of Messrs. Schöttler, Schröter, and Stodola;

b Reply of Professor Riedler, all in the issue of February 25, 1905;

c General symposium upon the mechanical efficiency and indicated power of the gas engine, including contributions from Messrs. Stodola, Riedler, Schöttler, Meyer, Ehrhardt and Wagner; issue of April 1, 1905;

d Communication from Mr. Rudolph Diesel upon the mechanical efficiency and indicated power of the gas engine; issue of May 20, 1905;

e Communication from Professor Hugo Guldner; issue of June 24, 1905.

DISCUSSION

THE CONVEYING OF MATERIALS

DISCUSSION OF PAPERS GIVEN AT DETROIT

At the Detroit meeting the second session was devoted to a symposium on the hoisting and conveying of materials and five papers were presented as follows:

Hoisting and Conveying Machinery, G. E. Titcomb.
Continuous Conveying of Materials, S. B. Peck.
The Belt Conveyor, C. Kemble Baldwin.
Conveying Machinery in a Cement Plant, C. J. Tomlinson.
Performance of Belt Conveyors, E. J. Haddock.

The first four papers were published in June Proceedings and the last paper, by Mr. E. J. Haddock, appears in this number. The discussion upon these five papers here follows complete.

Mr. Spencer Miller Mr. Titcomb briefly refers to cableways, showing that he regards such as hoisting and conveying machines. This is my excuse for adding something to his paper on the subject of cableways.

2 By a cableway I mean a hoisting and conveying machine employing a suspended cable as a track-way. When the hoisting function is omitted we call the device a wire rope tramway. There are about 1000 cableways in the United States. The field covered by the cableway is so extensive and the variety of cableways so great, that the subject is worthy of a separate paper. The loads handled by cableways range from one-half ton to 25 tons. Fifty-ton cableways have been designed, but so far as I know, have not been constructed. The length of the cableway span varies from 200 ft. to 2500 ft.

3 In the matter of speed, the load carriage travels usually from 600 to 1500 ft. per minute, while there are instances of cableways, especially those used in coaling warships at sea, where the carriage speed is over 3000 ft. per minute. At the Lidgerwood cableway

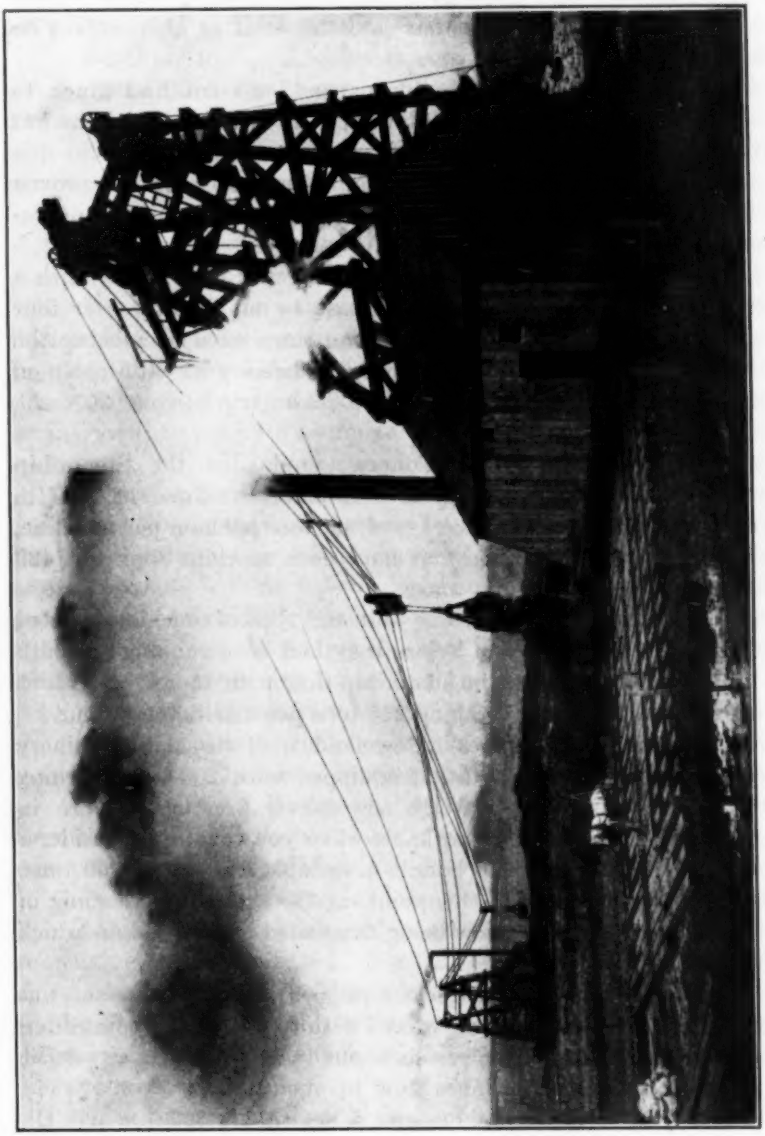
testing station, speeds of 3600 ft. per min. have been accomplished and apparently such speeds are entirely practical for light loads. Cableways therefore seem to outdistance bridge tramways in the matter of speed and span. There are few problems solvable with the bridge tramway that are not solvable with a cableway. Cableways have been built with "man trolleys" which eliminate the long hoisting and conveying ropes, with their attendant fall rope carriers.

4 Recent inventions properly applied multiply the life of the main cable and remove any objection on the standpoint of repairs. Cableways are successfully and economically employed in unloading ships at the wharf, operating grab buckets, and operating scraper buckets. For construction of dams and locks they stand supreme. As excavators of canals they are frequently the most economical machine to employ. The most economical method of employing cableways in canal excavation is by the use of what is known as the duplex cableway. Two double towers placed on wheels are movable by power, lengthwise of the canal. Two cableways cross the canal. Grab buckets are handled by the cableways. Spoil banks are on both sides of the canal. Material is excavated by one cableway and carried speedily towards one tower, while by the other cableway it is carried in the opposite direction. Hence the bucket travels only half the distance, and therefore makes many more trips than a single cableway which has to do the entire work.

5 Many cableways are operated by electric motors both with the alternating and direct currents.

6 The greatest contest for economy of canal construction was witnessed in the excavation of the Chicago Drainage Canal, where the two leading machines were a bridge tramway and a traveling cableway. When all costs were taken into consideration, repairs, interest on plant, etc., the cableway won by several cents per cubic yard. It may be truthfully said that the cableway with its towers traveling on wheels was born on the Chicago Drainage Canal. This was also the first great construction work where the bridge tramway was employed. It would be interesting to know if a bridge tramway has been employed on construction work since. The great field of usefulness in bridge tramways has been in ship building, unloading ore and coal, and serving as a material handling machine in manufacturers' plants.

Mr. Melvin Pattison Referring to the paper on Hoisting and Conveying Machinery, Mr. Titcomb, has given the type of ore-



DUPLEX CABLEWAYS

handling machinery that is most in use very little mention; and while he refers to some very interesting coal-handling equipments, he does not mention at all the type that ranks at least as high as any to which he refers.

2 In discussing Mr. Titcomb's paper, I do not find much to criticise in what he has said, but what I do criticise is that he has left so much unsaid.

3 Fig. 4 of his paper shows a type of ore-unloader that has proven to be the most efficient grab bucket five-ton unit machine in operation for taking ore from vessels.

4 Each machine is equipped with an electric man trolley with a turn-table. The operator can with ease be making three or four distinct and different operations at one time, such as racking the trolley, turning the bucket and hoisting or lowering it. It is common practice for these machines to make a round trip in from 30 to 35 seconds.

5 The four machines at Conneaut unloaded the Steamship Hoover & Mason with 9202 gross tons of ore on June 14, 1907, in 7 hr. 12 min., which was an average of 326 tons per hour per machine, including cleaning up. The first hour each machine took out 429 tons, and the last hour 150 tons.

6 There were six of this same unit and type of machines started last season at Fairport, and before they had been running a month and a half they unloaded the steamship Cole with 11 385 gross tons of ore in 8 hr. 58 min., averaging 215 tons per machine per hour.

7 There are four machines in Cleveland, with the same machinery part equipment, but in addition equipped with 200-ton weighing bins, which can accurately weigh and deliver a carload of ore in about 7 sec. This seems remarkable when you take into consideration that the scales have to handle a variable load up to 250 tons. I call especial attention to this point, as the successful weighing of ore from vessels to cars is now being demanded by railroad and dock companies.

8 There are many other docks equipped with this type of unloaders; and the summary of what I wish to say in this connection is that they take ore out of vessels about twice as fast as any other type of five-ton unit machines now in operation.

9 Referring to Fig. 5, the bridges, as shown, are 565 feet over all, and are equipped with man trolleys, but do not have the turn-table feature.

10 The first bridge built operates a $7\frac{1}{2}$ ton grab bucket, and will

handle 350 tons of ore per hour from the end of the cantilever to the middle of the bridge. The second, or last bridge, operates a ten-ton grab bucket and will handle 500 tons per hour.

11 Referring to Fig. 6, under the heading of blast furnaces I do not recognize the bridge shown, nor do I understand why this bridge is chosen when the majority of large blast furnace plants are equipped with bridges of the same make as the unloaders referred to above, to say nothing of the parabolic bins, larries, transfer cars and furnace hoists, which means that this machinery handles a large portion of the iron ore from the time it leaves the vessel until converted into pig iron.

12 Referring to the handling of coal, I note there are eight illustrations of anthracite-handling machinery, and only six of bituminous. In 1907 there were 375 000 000 tons of bituminous coal mined in this country, and only 67 000 000 tons of anthracite coal, which means less than one-fifth as much anthracite coal handled as bituminous.

13 There seems to be quite a change in sentiment among coal dock managers in the last three or four years, since four years ago the larger and more important concerns believed in small unit dock hoists, breaking bulk on face of dock, and delivering into stock pile by means of cars or trolleys, the dock people now seem to be in favor of larger units, and the bucket that goes into a vessel delivering into stock pile, thereby saving rehandling and breakage.

14 Further, sentiment seems to be growing in favor of man-trolley equipments, similar to what has become the standard for handling iron ore. In fact, coal is now being handled with man-trolley equipments at New Haven, Worcester, Springfield, Boston, Charleston and Superior, as well as at other places, and there is a bridge of this type now being built in New York, which will be referred to later.

15 Some of the advantages of the man-trolley are that the operator is always with his work, and long leads of whipping lines, sheave supports, etc., are eliminated. Fig. 12 and Fig. 13 refer to an anthracite coal-handling and storage plant at Superior, which is about one-quarter of the total problem on this dock, the other three-quarters being the handling of bituminous coal, which is not mentioned.

16 The bituminous dock is equipped with what is probably the most flexible coal-handling machinery of the intermittent type now in operation anywhere.

17 The equipment consists of four electric unloading towers on

the face of the dock; a line of pockets immediately back of the same, with three electric transfer cars on top of the pockets and back of the towers; four three-span bridges at the rear of the pockets and extending the width of the dock which is 500 ft., and a line of pockets on the rear of the dock running parallel with the pockets on the front.

18 The bridges are equipped with special man trolleys of two styles, one for stocking coal and one for taking coal out of stock with shovel buckets.

19 The fast plants will unload vessels at an average rate of about 150 tons per hour per machine, including cleaning up, and there are always one or two bridges available to fill all current orders from day to day for the different kinds of coal that are not being unloaded from the vessel.

20 Another plant of this unloading type, is the one in operation at the National Tube Company's Works at McKeesport, Pa., which is one of the fastest of which the writer has heard.

21 Another type is the Milwaukeee Coke and Gas Company's two machines at Milwaukee. These are steam driven and operate a two-ton bucket. On June 9 and 10 of this year they unloaded the steamer Powell Stackhouse with 10 002 tons of coal, and averaged, without taking out any delays, 175½ tons per machine per hour, including the cleaning up of the vessel with the grab buckets.

22 These two machines were in operation just seven months last season, and one-third of this time there were no vessels at the dock. The machines were actually working four months and twenty-two days, and unloaded in that time 586 000 tons of coal, which is best the record I know of for any dock.

23 Referring to Fig. 21, this is one of the modern rope-driven trolley bridge plants operating a three-ton grab bucket, and I wish to call attention to the Pittsburg Coal Company's No. 6 Dock at Superior, erected four years ago, and which handles only 1¼ ton buckets.

24 The general scheme of the two docks is similar, but the No. 6 Dock with only 1¼ ton buckets has a record of unloading a vessel at the rate of 108 tons per hour per machine, including cleaning up, delivering the coal at the rear end of the cantilever, which is 290 ft. from the apron, while the best record given for the machines shown in Fig. 21 is 135 tons per hour per machine with three-ton buckets. This No. 6 dock is steam driven, and has the same rope system as the Milwaukee dock which was referred to above.

25 While these two steam driven, rope system bridge equipments are successful for handling bituminous coal, there is, as stated above, a growing sentiment in favor of man trolley and larger unit grab buckets, which is very strikingly illustrated in the new bridge which is being built for the Astoria Light Heat and Power Company, at Astoria, Long Island, which is a part of the Consolidated Gas Company of New York.

26 This bridge is 603 ft. over all, and is equipped with a man trolley, handling a grab bucket that will take between six and seven tons at a grab.

27 The moving load on the bridge is 40 tons. The speeds are:

	Feet per minute
Hoisting	225
Trolley travel	1200
Bridge travel	75

28 This bridge will be ready for operation the latter part of July, and will be the largest coal-handling bridge crane.

Mr. George B. Willcox Salt and chemicals are noted in Par. 75 of Mr. Baldwin's paper, as substances that can be satisfactorily handled with belts, but the kind of belts is not specified. In the salt machinery business our experience has been that belts are only suitable for certain kinds of service as salt carriers.

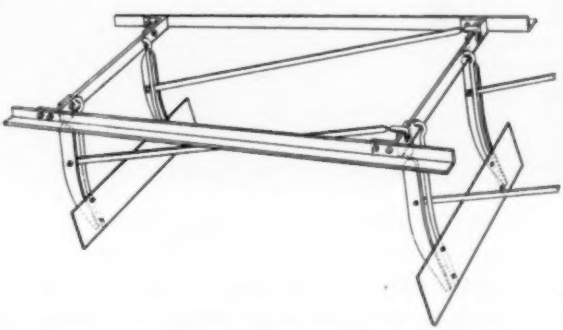


Fig. 1 Reciprocating Scraper Conveyor

2 For loading vessels in bulk, a belt conveyor located in the warehouse floor and delivering into a chute high over the wharf and discharging in the hold of the ship is satisfactory.

3 With 30 men wheeling salt in barrels and dumping at both sides of the belt, a 22-in. belt has delivered 150 tons per hour into the hold of a vessel, and with 25 men an average run is a cargo of 1100 tons in 10 hr. When navigation closes, the belt is cleaned and removed from the pulleys. For such service, namely, where a large quantity of salt is handled in a short time, probably nothing is so satisfactory as a belt. But where it is required to run

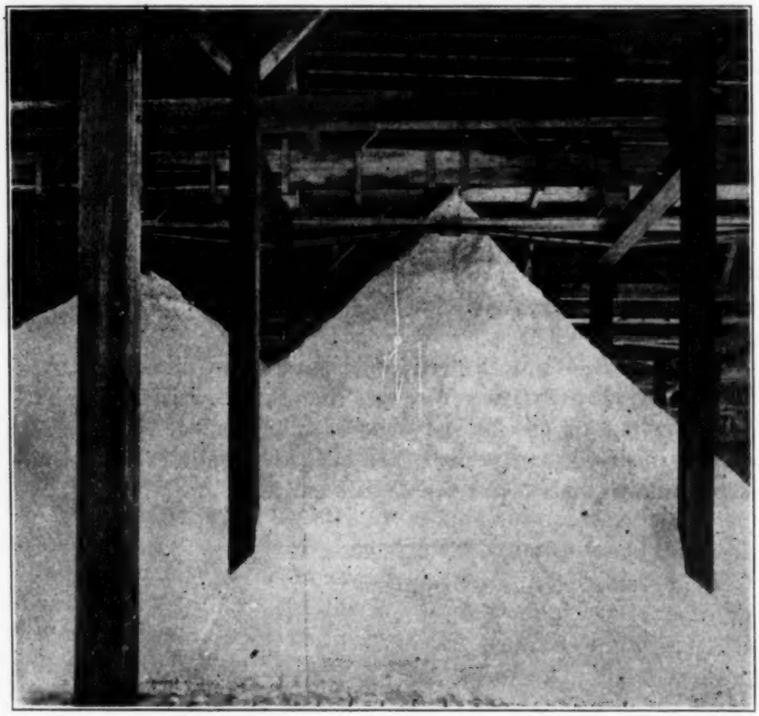


FIG. 2 CONVEYORS IN A SALT PLANT

continuously, carrying only a small stream of salt, a belt is far from satisfactory. The state of the weather affects the operation. On a dry day newly made salt may discharge well from the belt surface, but a cold damp day causes it to stick to the belt, and the belt surface becomes wet and slimy.

4 Brushes and angle scrapers of many kinds, including wood and plate glass, have been used on the belt to overcome this difficulty,

but not successfully. A serious fault is that atmospheric dust accumulates on the belt surface, which travels through the air many miles a day, and in consequence rings of dirty salt accumulate around the rollers at the edges of the belt, finally dropping on the salt pile. Such dirt spots through the salt are very conspicuous and seriously affect the market value of the salt.

5 We have abandoned belts entirely for continuous service of the class described, as for instance, removing the salt from a battery of grainers as fast as it is produced and distributing it in piles on the warehouse floor.

6 In such places the most satisfactory conveyor is of the reciprocating scraper variety, as shown in the accompanying illustration, Fig. 1. The side angles ride back and forth on slides made of 2 by $\frac{1}{4}$ angle about 8 in. long fixed to the side of the conveyor trough. The bottom of the trough is lined with $\frac{1}{2}$ in. unfinished plate glass which can be had from any plate glass factory at about 25 cents per square foot.

7 These conveyors work back and forth very slowly. The slides do not wear rapidly and are easily replaced. Five conveyors of this variety have been in continuous operation night and day for nearly two years without repair and with no perceptible wear. They take the salt from 10 grainers, at the rate of 1000 bbl. per 24 hr., and deliver it in 20-ft. piles on the warehouse floor. Fig. 2 is from a photograph of one of them. One of these salt piles as shown is about 22 ft. high.

MR. CHARLES PIEZ In giving his brief history of the development of the belt conveyor, Mr. Baldwin did not mention the introduction of what was known as the Robin's patent belt some twelve years ago. This belt combined, according to the claims of the company exploiting it, "the advantage of a central thickened wear-resisting cover with stiffened edges, making the belt bend more easily at the center and better preserving its troughed shape between the sets of idler pulleys. This stiffening is done by running two or three plies of duck a part of the way in from the edges."

2 The purpose the inventor had in mind was evidently to give to a belt a largely increased carrying capacity by forcing it to assume the shape of a deep, continuous trough. To accomplish this, idlers of the form shown in Fig. 7 of Mr. Baldwin's paper were used, but with troughing pulleys arranged at angles fully 15 deg. steeper than shown in this cut. Large carrying capacities for narrow widths of belt, however, carried with them certain serious defects.

3 In order to secure flexibility the strength of the belt at the center was sacrificed by omitting some of the plies of duck. The large increase in load per unit area of belt required a heavy increase in the tractive force necessary to drive the belt and this was obtainable owing to the limited width of the belt, only by largely increasing the initial tension. The resultant unit strain exerted on a belt that was defective in structure quickly brought about a destruction of the bond between the fabric and the rubber coating and this destruction was effectually assisted by the lateral bending to which the troughing pulleys subjected the belt.

4 The remedy, of course, lay in materially reducing the depth of troughing, resulting in wider belts for the same carrying capacities, a much smaller degree of lateral flexure from the troughing rolls, lower unit stresses per unit width of belt and the advantage of using belts of uniform ply throughout their width. The progress of this development is shown in Mr. Baldwin's illustrations.

5 The original idler used for the patent or flexible belt was of the three-pulley type shown in Fig. 7 but with troughing rolls arranged at angles of about 40 deg. The next step was the reduction of the angle of the troughing rolls, some makers using 20 deg. and others 25 deg.

6 Fig. 5 of Mr. Baldwin's paper shows a concave roller developed by one of the users of a wide flexible belt for the purpose of overcoming the destruction of the belt due to the severe lateral flexure of the troughing rolls. The aim was to approximate in the idler the catenary curve that the belt would assume if supported only at the outer edges.

7 Fig. 6 of Mr. Baldwin's paper shows the unit troughing idlers arranged in substantially the same way, resulting in giving the belt practically a uniform curve in a transverse section. This form of idler has been adopted by a number of the leading manufacturers for the wider belt conveyors, and as is readily apparent makes possible the use of a standard belt of uniform ply. This to my mind is a distinct advantage of the shallow trough belt conveyor.

8 The function of the belt in belt conveyors is twofold. First, it serves as the carrier of the material; second, it serves as the transmitting medium for the power necessary to move the material. The second function is undoubtedly best performed by a belt of uniform strength throughout its width; and if for purposes of carrying the material a form of troughing is adopted to which a uniform ply belt will readily accommodate itself, then there can be no question as to the best structure of belt to use for conveying purposes.

9 But while the shallow trough belt with idlers such as shown in Fig. 6 and 7 has practically superseded the deeply troughed belt, there is at present a noticeable tendency to revert to the old type of flat belt. This tendency is illustrated by the increasing use for certain purposes of the idler shown in Mr. Peck's paper, Fig. 9. This idler is flat through the greater part of its length, the two end sections being slightly convex to confine the material. In actual practice the end diameter is about 1½ in. larger than the center diameter. The pulleys are very light, being made of pressed steel, and the shafts run in pivoted, chain-oiling bearings. The belts used with them are of uniform ply, usually protected with an extra thickness of rubber on the carrying side. For pulverulent, granular or sized material not exceeding 2½ in. in diameter, and these classes, let me state, represent in volume the greater part of the material carried by belts, the slight troughing given by this idler is sufficient to retain the material on the belt.

10 The only charge that can be made against it, according to the standard set by Mr. Baldwin, is that the slip incidental to the difference in peripheral speeds of two portions of the idler is necessarily fatal to durability in the belt. But are the idlers of the construction shown in Fig. 6 and 7 of his paper free from slip? These idlers must be lubricated with grease because, as Mr. Baldwin very properly points out, oil would be hard to confine and contact with the rubber would cause rapid deterioration. But grease is if anything a retardant for very light loads and with the idlers spaced as closely as they are on the carrying side, the pressure per inch of journal bearing is small, particularly when the belt is running lightly loaded. Then, too, there is the friction due to the end pressure of the hubs of the inclined pulleys against the bearings. With such conditions existing, it is only natural to suppose that some slip will occur and the evidences of it are easily discernible to the practiced eye, for it is very rare to find all rollers on an installation running free when the belt is running light, and the idlers quickly become polished on the surface.

11 There is also the fact that a horizontal conveyor running light absorbs almost as much power as when fully loaded, the no load readings frequently being from 90 to 95 per cent of the full load readings. This slight difference can only be explained on the assumption of proportionately lower journal friction under full load conditions than under conditions of light loading. The inference is therefore that the idlers lag considerably under a light load and it is only fair to assume that they continue to lag to some extent under

average load conditions. For after all, large capacity is secured with belts by carrying light loads at high speeds, and the average carrying loads are insufficient to operate grease lubricated journals at full efficiency.

12 Slip is therefore present in some degree in all idlers, and it is my opinion that with their greater lightness, their more perfect lubrication, and their better alignment, the idlers in Fig. 9 of Mr. Peck's article are no more destructive to the under side of the belt than are those illustrated by Fig. 6 and 7 of the article under discussion. They are immeasurably kinder to the belt, however, in avoiding lateral flexure and on this account add very much to its durability.

13 It is almost an axiom in the belt conveyor art that belts wear out on the carrying side. Eliminating from consideration the deterioration caused by troughing, the reasons for the greater wear on the carrying side are:

a Impact of the delivered material;
b Action of pulleys of too small a diameter on the belt;
c Improper location and design of driving mechanism.

14 Mr. Baldwin has very fully covered the first cause but it may be interesting to add that where a mixture of fine and coarse material is handled, a very effective means of reducing the destructive action at the delivery point consists in inserting a short screen near the delivery end of the chute, thus permitting the fine stuff to sift through and form a cushion to receive the impact of the heavier pieces.

15 In regard to the second cause of the deterioration of belts altogether too little value is placed upon the use of pulleys of the proper diameter. I do not think it was Mr. Baldwin's intention to make it appear that the diameter of the driving pulley depends only on the width of the belt, yet that is the inference one unconsciously draws from Table 1. He does mention in another portion of his paper the minimum number of plies for various widths; but the relation between number of plies and diameter of driving pulley should be given in the table, for it is the number of plies and not the width of the belt that determines the size of the pulley. The action of a belt on a pulley is similar to that of a rope on a sheave and when the belt is forced around a pulley of too small a diameter, the outer layers of canvas stretch so materially with relation to the inner layers as to bring about a gradual destruction of the band between them.

16 While Mr. Haddock's experiments seem to indicate that there is only a slight gain in tractive effect when the pulleys exceed in

diameter 5 in. for every ply; that is, that there is no great gain in driving power in a 30-in. pulley over a 20-in. pulley for a four-ply belt, yet these experiments touch only one side of the problem, and do not take into consideration the durability of the belt.

17 I feel most strongly that driving pulleys should have not less than 8 in. in diameter for every ply of canvas, and that no pulley around which the belt passes under full load should be of smaller dimensions.

18 Take Fig. 10 of Mr. Baldwin's paper for instance, the head pulley appears to be about two-thirds the diameter of the driving pulley. Assuming the belt to be 30 in. six-ply the layman would conclude from the table in Par. 41 that good practice would make the diameter of the driving pulley 30 in. and of the head pulley 20 in. Yet this combination would be a destructive one, for the belt is under as heavy a strain bending around the head pulley as it is on the driving pulley.

19 Now as to the third cause of deterioration, the improper location and design of the driving mechanism.

20 There is a fundamental principle which applies to all transmission media whether they be chain, rope or belt. It is this: make as few turns or bends under full stress as possible and avoid counter bends.

21 With this principle in mind there is only one proper position for the driving pulley and that is at the head or discharge end of the conveyor. In such a drive as shown in Fig, 10 of Mr. Baldwin's paper, the compound drive, using pulleys of as small diameter as mentioned in Pars. 38 and 39, might be mentioned as expedients but should hardly be cited as illustrative of sound practice. The conveying art, rapid and brilliant as its progress has been, is replete with examples of expediency, and it is now high time that consideration should be given to cost of maintenance and operation as well as to initial cost.

22 Let us now turn to Mr. Baldwin's formula for the horse power required for belt conveyors. The formula for horizontal conveyors

$$\text{h.p.} = \frac{C \times T \times L}{1000}$$

given by him suggests that the power is directly proportional to the load in tons per hour, yet that is not the case. For to get even approximate results with the formula the factor T should be a constant as well as C and Mr. Baldwin might have combined to advantage $C \times T$ into a single constant and given a table of values for it.

23 Some readings of belt installations are as follows:

a A 30 in. belt conveyor handling crushed bituminous coal. Conveyor is horizontal with 253 ft. centers. Speed 600 ft. per minute. Load carried at time of test 215 tons per hour. Conveyor is provided with self propelling tripper. Readings, including motor and drive, are as follows:

	Horse power
Starting load empty	24.3
Running empty	11
Running loaded	11.7
Moving tripper	13.3

Belt empty

Mr. Baldwin's formula gives 9.18 h.p. without tripper.

Mr. Peck's table gives 11.70 h.p. without tripper.

b A 30-in. conveyor, 386 ft. centers, running horizontally at 348 ft. per min., carries coke at the rate of 60 tons per hour.

The readings including motor and drive were as follows:

	Horse power
Empty	13.4
Loaded	13.4 to 14

Mr. Peck's table shows, with an allowance of 10 per cent per drive but without considering the motor efficiency, 11.36 h.p.

Mr. Baldwin's formula, substituting 60 tons for the capacity in tons per hour, gives 3.86 h.p.

24 Of course Mr. Baldwin states that the power should always be figured for the full capacity at the chosen speed but what is the full capacity of coke on a 30-in. belt at a speed of 348 ft. per minute.

25 I am dwelling on this point to indicate that while the formula may yield results in the hands of a man who is privy to its peculiarities, it is not a good formula for general use.

Mr. T. A. Bennett With reference to Fig. 9 in Mr. Peck's paper and his mention of the four-roll idler which he considers best for modern practice, I would call attention to the so-called five-pulley idler shown in illustration herewith and mentioned in Fig. 6 of Mr. Baldwin's paper. This latest development of the idler form avoids

the crease at the center of the belt where the greatest load comes. It conforms to the natural curve of the belt and retains the valuable feature of a center horizontal pulley. This horizontal pulley is necessary for guiding purposes, as without it the belt rides up and down the inclined idlers according to its loading.

2 With reference to belts Mr. Baldwin says truly in Par. 20 that "the belt conveyor industry has been built up mainly on the remarkable showing of the rubber belts made by Mr. Robins." I expected to have to call attention to the absence of the patent belt in Mr. Baldwin's paper, but I see this omission has been well brought out by preceding discussion.

3 It was found that the requirements of a conveyor belt were:

a extra rubber cover at the center where the greatest wear comes;

b pliability to fit the curve of the troughing idler;

c heavy edges to stiffen and strengthen it.

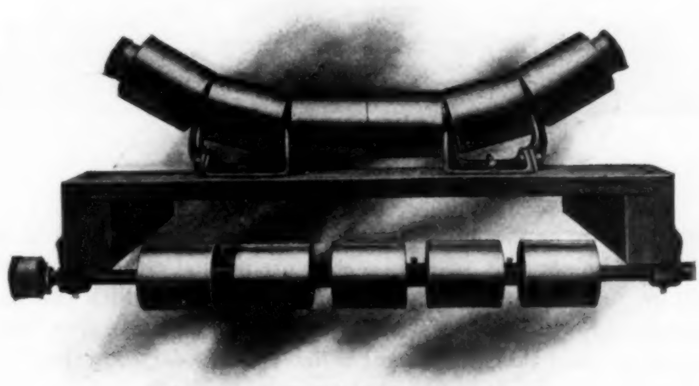


FIG. 1 FIVE PULLEY IDLER

4 I desire to correct the statement that the wear on the cover is uniform. We all know that the wear comes mostly in the center, the plies wearing away in steps rising to the edge, which is often not worn. These requirements were all met in practice by the patent belt made by the Robins Conveying Belt Company, a belt of uniform thickness, with a heavy rubber cover and extra plies of duck extending in part way from

each edge, thus protecting and strengthening the edges and saving the use of unnecessary rubber. The resulting belt can be made as heavy at the center as will conform to the troughing idler and the heavy duck edges add stiffness and strength to the belt.

5 Regarding the waterproof qualities of rubber belts referred to in Par. 27, while it may be correct to state that they are not "absolutely waterproof," they are practically so. The friction rubber is forced into the mesh of the duck and not only surrounds each thread but covers each ply, insulating it from its neighbor. An investigation of many belts after service has shown me that the moisture from a cut or puncture extends around it into the belt only about an inch; in fact we all know that the superiority of rubber belts in the resistance of moisture was the cause of their original adoption as driving belts in damp places.

6 Rubber belts with extra cover are the only ones that can stand the service mentioned in Par. 27 in handling dredgings, wet tailings, etc. On over fifty gold dredges in California, the Yukon and else-

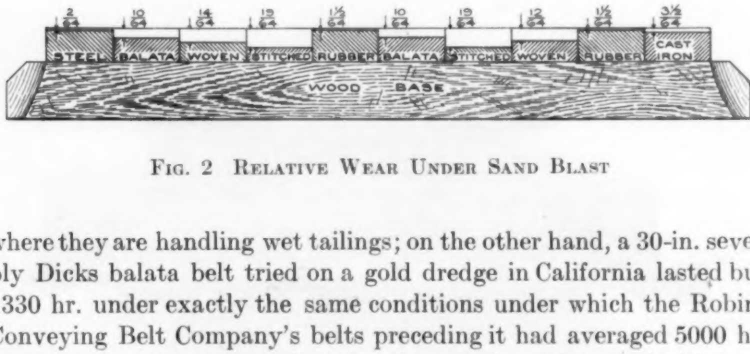


Fig. 2 Relative Wear Under Sand Blast

where they are handling wet tailings; on the other hand, a 30-in. seven ply Dicks balata belt tried on a gold dredge in California lasted but 1330 hr. under exactly the same conditions under which the Robins Conveying Belt Company's belts preceding it had averaged 5000 hr. The question of moisture is not a determining factor in the life of a well-made rubber belt except at the very end of its service.

7 The accompanying cuts show the relative ability of various belts and materials to resist abrasion. On a board are mounted the following materials, named in order from left to right: Cast iron, rubber belt, woven cotton belt, stitched canvas belt, balata belt, rubber belt, stitched canvas belt, woven cotton belt, balata belt, bar steel.

8 This block was subjected to the action of the sand blast by passing the nozzle uniformly back and forth over the center line for a period of 45 min. From this and many past experiments, we find

the relative abrasive resisting qualities of these materials to be as follows, taking the volume of rubber belt worn away as being 1.0:

Rubber belt	1.0
Rolled steel bar	1.5
Cast iron	3.5
Balata belt, including gum cover	5.0
Woven cotton belt, high grade	6.5
Stitched duck, high grade	8.0
Woven cotton belt, low grade	9.0 to 15.0

9 Par. 30 of Mr. Baldwin's paper will bear correction regarding impairment in the strength and vitality of the fabric in rubber belts by the great heat of vulcanization. The "great heat" mentioned is only that due to a steam pressure of 40 lb. or about 290 deg. fahr., which, of course, is not enough to affect the cotton.

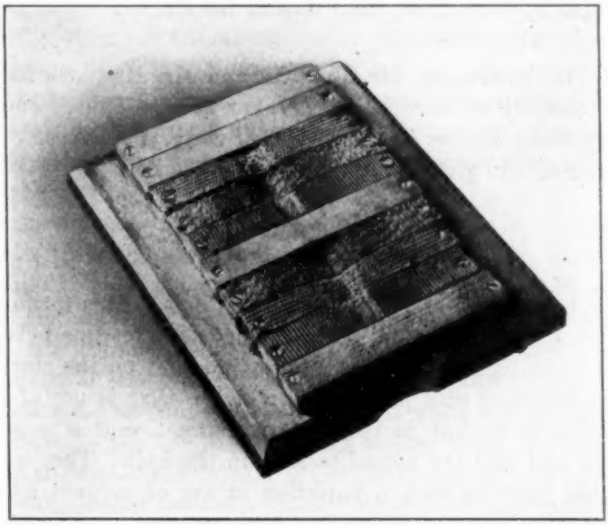


Fig. 3 Block Showing Results of Abrasive Tests on Different Materials

10 In Par. 42 appears a statement regarding the use of take-ups "to adjust the belt should the splice not be exactly square." Such a remedy would make the belt run like a screw thread to one side and would start the belt away from the take-up pulley at an angle with the center line of the conveyor. In one part of the belt one edge would be longer than the other, which would tend to make the belt run to

one side and off the troughing idlers. Evidently the best practice is to keep the take-up shaft square with the center line of the conveyor and to remedy the splices by making them square. It is very important that the take-up shaft be square with the axis of the conveyor.

11 Par. 43 brings up the detail of the discharge from the belt. This discharge follows the path of a trajectory based on the velocity of the material and independent of the diameter of the pulley. The course of its fall is modified somewhat if the material clings to the belt: For clinging materials a clean discharge can sometimes be gotten only by high speed.

12 While there are mechanical difficulties to be overcome connected with the belts, drives, idlers, etc., the real engineering in the conveyor business is in the handling of the materials themselves. The success of an installation depends upon an intimate knowledge of these materials and of their chemical and adhesive action on belts and most important, their flow in the chutes.

Mr. E. H. Messiter Mr. Baldwin and Mr. Haddock have dealt with the subject of tension in conveyor belts. This is one of the most important factors in the design of conveyor drives.

2 Mr. Baldwin gives for the tension in the belt

$$\frac{\text{h.p.} \times 33\ 000}{S \times W}$$

S is the belt speed and W the width of belt. This formula, however, gives the effective pull per inch, or the difference between the tension on the tight side of the pulley and that on the slack side.

3 The effective pull per inch is a figure that is useful in power transmission work, but in belt conveyor practice it is necessary to go further and find the actual tension in the belt. The reason for this is that there is such a variation in arc of contact and in the coefficient of friction of the pulley surfaces used in different drives that the relation between tension and effective pull varies widely.

4 To find the actual tension we resort to the old formula

$$\frac{T_1}{T_2} = 10^{0.000758 f \alpha}$$

which means that the logarithm of the ratio between the tensions on the tight and slack sides is equal to 0.000758 multiplied by the product of the coefficient of friction into the arc of contact.

5 Combining the above equation with the following formula for the effective pull, we are able to find the tension in any conveyor belt.

$$T_1 - T_2 = \frac{\text{h.p.} \times 33\,000}{S}$$

$$\text{Putting } \frac{T_1}{T_2} = 10^{0.000758 f \alpha} = R$$

we have for the maximum tension in a horizontal belt

$$T_1 = \frac{\text{h.p.} \times 33\,000 \times R}{S \times (R-1)}$$

6 In regard to the belt conveyor with shallow trough which has been mentioned by Mr. Peck and Mr. Pease a question suggests itself.

7 Remembering that the earliest conveyor belts were flat and that the only object of troughing was to gain an increase of capacity so that when a belt has to be renewed it will cost less money, it may be asked: unless we are going to get a substantial increase of capacity by means of a good deep trough, what is the use of troughing at all, and thereby sacrificing the simplicity of the single roller that suffices to carry the flat belt?

8 In connection with the subject of the diameter of conveyor belt pulleys, Mr. Haddock has proposed a minimum of 5 in. for each ply of belt while Mr. Pease contends that 8 in. should be the limit. The argument of the latter by analogy from transmission belts and driving ropes fails to take account of the wear on the face of the belt due to the impact of the material carried. In a properly constructed belt conveyor for anything but the lightest work the life of the belt is determined by the length of time its rubber cover withstands the abrasion of the material carried. We can therefore take some liberties in the way of smaller pulleys and counterbinding and can, I believe, use something not far from Mr. Haddock's recommendation for the minimum diameter of pulleys. Conveyor belts are certainly made today that wear out, as they should, by the abrasion of the rubber cover.

9 Passing to the question of the power consumption of belt conveyors, I agree with Mr. Pease that it would be desirable to have a uniform method of computation, but it will never be possible to test the constants for a conveyor of one make by trial with a conveyor of another design.

10 Mr. Baldwin has given a formula for horse power in which the constant is multiplied by the capacity and the length without direct reference to the belt speed. In its original form this formula has been in use for several years. It is as follows:

$$\text{h.p.} = \frac{1}{1000}\left[L\,(F_L\,T + F_s\,S) + H\,T\right]$$

in which

L = Length of conveyor.
F_L = Load factor.
F_s = Speed factor.
T = Capacity in tons per hour.
S = Belt speed.
H = Height through which load is lifted.

11 This formula is applicable to any belt conveyor and will show the difference in power consumption between a loaded conveyor and one running empty.

MR. JAMES M. DODGE I have only a word to say. I was naturally much interested in Mr. Peck's remarks, inasmuch as Mr. Peck and I are business associates. He spoke very handsomely about a certain equalizing gear which I designed and which has been used with our elevators and conveyors. In this connection I desire to say that I was led to design this driving apparatus because Mr. C. W. Hunt, a competitor of ours, had previously designed an equalizing and driving mechanism and, consequently, I made my effort for competitive reasons. Mr. Hunt's apparatus was eminently satisfactory and if we could have used his device the invention of my device would have been quite unnecessary.

2 Mr. Peck also stated that it was desirable to make machinery for the handling of materials as well as it could be done, and it affords me pleasure to take this occasion to say that Mr. Hunt has always done the best he knew how, and we have tried our best to emulate him.

PROF. R. C. CARPENTER I merely want to say a few words on the subject of the paper of Mr. Tomlinson and on the subject of conveying machinery in the Portland cement industries. His paper gives the impression which I think to be an error, that belt conveyors are mostly used in these plants. He says that this industry more than any other depends upon conveying machinery for transporting

raw material from where it is mined to the mill and through all the processes required to manufacture it and to the bins for the finished product. I think the history of conveying machinery in the Portland cement plants shows that it has been the source of more trouble and expense than almost anything else connected with the industry. The material to be handled is very gritty and heavy; consequently it is hard upon the conveying machinery. The cost of maintenance may be entirely out of proportion to the cost of installation if light conveyors are installed; for that reason it pays in the end to put in very heavy machines. I think Mr. Dodge will bear me out in this, for I know that he has been called upon very often to repair defects in the conveyor system of cement mills.

2 I understand from Mr. Tomlinson's paper that the screw conveyor is used but little in cement mills. My experience is very different from this; my observation indicates that it is largely used in cement mills. If well built it is very satisfactory and will stand up to its work better than any other type; its field however is limited.

3 It cannot handle coarse stuff satisfactorily. The material must be broken up to a quarter-inch diameter or less in order to be handled satisfactorily by a screw conveyor. From my experience in cement mills I should say the belt conveyor would handle satisfactorily materials not dusty or gritty; but a large portion of the material in the cement industry is very dry and dusty and it is extremely difficult to prevent the belt from scattering a great deal of dust, for which reason the belt conveyor has failed in many cases. I might say, however, that there are many belt conveyors in satisfactory use in the cement industry although the proportion to the total is small.

4 One form of conveyor has not been mentioned here, I think, one originally designed and patented by Mr. C. W. Hunt which has proved satisfactory in the cement industry, a car system handled by a continuous moving steel rope. This is one of the most satisfactory systems of conveying that has been designed for the cement mills. If a car breaks down it does not stop the whole plant, as another can be supplied without stopping; whereas with other types if a part of one conveyor breaks down it may put out of service practically the whole plant.

5 Another word in regard to the elevator in cement mills. The material has to be handled and carried both horizontally and vertically, and so arranged that it can be moved over and from one level to another. Mr. Hunt introduced a system of automatic cars which may be pushed on elevators and carried up to different levels and

shoved over to the desired point by trolleys or by hand. That system is coming into almost universal use and is as well thought of as any system at the present, in my opinion.

MR. E. S. FICKES Mr. Peck in his very interesting paper does not describe a conveyor which has been found quite useful for handling finely divided dry materials and particularly those which are gritty and dusty. This conveyor consists of a tube with its axis either horizontal or slightly inclined; the tube is rotated on external bearings or trunnions and the material is carried forward either by the slope of the tube, if it is inclined, or by a helicoid or helicoidal arrangement of vanes or blades on the inside of the tube. The helicoid, or the vanes which replace it, revolves with the tube, thus carrying the material forward. As the material is inside the tube it causes no wear excepting that due to its flow as it rolls spirally through the revolving tube from one end to the other. The bearings and trunnions are not exposed at all to the material being conveyed.

2 When the conveyors are small they can be built of rolled steel pipe, with rolled or cast helicoids forced or keyed into them so that there will be no motion between the helicoid and the outer shell. Larger conveyor shells must be built up of plates with either a continuous helicoid or suitable vanes fastened to the shell. Excepting in the case of very small conveyors, trunnion bearings are used, the conveyor being carried preferably by a turned, rolled steel tire which rests on the trunnion bearings. The conveyors can be driven either by gears, sprocket chains, or belts.

3 When the material is transferred from one length of conveyor to another, or when it is necessary to put an angle in the conveyor, the end of the conveyor is fitted into a box or transfer station into which the material is discharged and from which it is fed into the second section.

4 A simple arrangement to prevent the escape of dust from these transfer stations consists of a series of outstanding circumferential ribs on the shell of the conveyor which mesh into a corresponding series of stationary ribs on the transfer station with just enough space between to prevent them from rubbing. This arrangement is particularly useful if the materials are very gritty, in which case the grooves on the transfer station must be designed to discharge into it the dust which lodges in them. When the material is not gritty some simple form of packing ring or stuffing box can be used, although its use will increase somewhat the power required to run the conveyor.

5 The material is fed into the smaller sizes of conveyors by suitably shaped blades attached to the open end of the tube. When the conveyors are large enough a better arrangement is to carry the feeding spout into the conveyor a short distance, the end of the conveyor being covered by a head which prevents the material from rolling out of it.

6 This type of conveyor is particularly useful in chemical works and similar plants where lime, soda ash and other materials are handled, the dust of which is often extremely disagreeable and annoying if allowed to escape. It is also a very useful conveyor when such dusty materials have to be heated or cooled as well as carried from place to place in the works. If the material is extremely hot, it is necessary either to line a portion of the conveyor with fire brick or use water jackets to prevent the heat from burning or otherwise injuring the machinery. When the material is not too hot, a cast iron shell can be used with cast iron helicoids or blades for the parts exposed to the greatest heat, these being made so as to be easily replaced when destroyed. The cooling of the material as it is conveyed can be accelerated by spraying water on the outside shell of the conveyor, suitable guards being placed to keep the water off the bearings and from getting into the transfer stations.

7 With some conveyors which require the material to be gently heated or maintained at a uniform temperature as it passes from one department of the works to another, a few gas jets placed along the under side of the conveyors are used. If, however, more accurate regulation of the temperature is required and the material does not need to be heated too highly, a stationary steam heated pipe placed on the axis of the conveyor can be used, the shell being covered with insulating material to decrease the loss of heat by radiation.

8 It is evident that this type of conveyor is not adapted to material which from dampness or other causes would tend to cling together and choke it, nor for the same reason can material containing large lumps be handled through the smaller sizes having a continuous helicoid which, with the shaft about which it is coiled, fills the entire tube.

9 The high first cost of this type of conveyor, as compared with its carrying capacity, greatly limits its field of usefulness, but for some materials and under certain conditions it is a very useful machine.

THE SPIRAL SPRING CONVEYOR BELT IDLER

Mr. E. G. Thomas A new form of troughing idler pulley for conveyor belts employs, instead of the usual cast iron pulleys, a flexible roller consisting of a spiral spring, 5 in. in diameter, rotatively supported in horizontally pivoted bearing boxes. The spring, under the weight of the belt and its load, stretches and sags to a smooth curve of approximately circular shape and the belt bends to a troughed

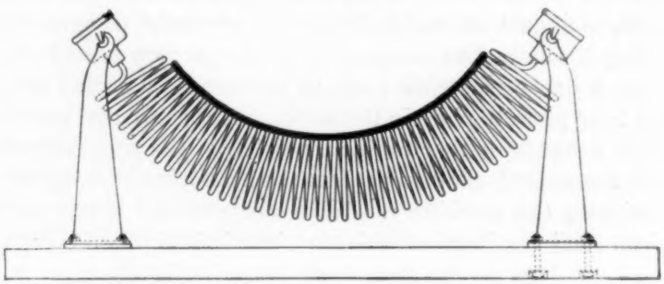


Fig. 1 Spiral Spring Roller for Belt

form in contact with and supported by the spring across its entire width. As the spring is of the same diameter throughout, there is pure rolling contact at all points between the spring and belt. The spring has about two coils per inch of belt width.

2 The size of wire used for the spring varies according to the load to be carried and the commercial idler will be so made that the extreme fiber stress will be about 15 000 lb. per square inch. As

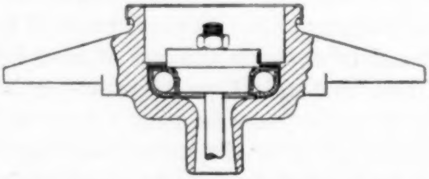


Fig. 2 Ball Bearing for End of Spring

the variation in fiber stress as the spring turns is only from one-fifth to one-tenth of the maximum, a satisfactory life for the spring is assured. In all cases the spring is designed to have such strength that when the belt is carrying its maximum load the area of the cross section of the load will be the same as that of the ordinary 35 deg. three-pulley idler, so that the two forms have equal carrying

capacity. Under any less load than the maximum the depression of the spring and the bending of the belt is less than the extreme.

3 The pivots of the bearing boxes permit them to accommodate themselves to the varying direction of the pull of the spring as it is

FIG. 3 LOADED CONVEYOR WITH SPIRAL SPRING IDLER

depressed more or less by the changes in load, and as the strain on the bearing is chiefly that of thrust, ball bearings are used to advantage.

FIG. 4 BELT RUNNING EMPTY

4 By the use of this idler, the curvature of the belt, instead of being concentrated along certain longitudinal lines, is equally distributed across its entire width and the degree of curvature is so small that wear from this source is practically eliminated. The

weight of the spring is much less than that of corresponding cast pulleys and the power required for its rotation in ball bearings is much smaller than is needed for the usual flat bearing with grease lubrication. The cost of lubrication will be reduced to the use of sufficient light oil at long intervals to keep the bearings free from rust. In the conveying of material containing large masses, such as the tailings conveyors of dredges, the spiral spring idler has important advantages, and because of its elastic character, will receive no damage from the shock of a falling rock which may break the rim of a cast pulley.

5 In the illustrations, Fig. 1 is an elevation of the idler under full load, Fig. 2 is a cross section of the bearing box, and Fig. 3 and Fig. 4 are photographs of the idler in use under a belt carrying coal.

Prof. H. Wade Hibbard I believe that the papers and discussion upon this subject have thus far come wholly from the makers or promoters of conveying machinery. I have no doubt but that the present and prospective users of such apparatus will receive the greatest profit from this meeting. Perhaps it may not be amiss to add a few comments from the point of view of one section of users, the railroads. The growth of the transportation business in this country has been so marvelous that changes have taken place almost under our eyes without being seen or without due recognition of their importance. The section of transportation concerned with this subject is the locomotive terminal. It is not so long since it was known as the roundhouse. But as transportation has enlarged, a new problem has arisen, or rather an ever-existing problem has taken on greatly enlarged importance. Trains have to run; today we are realizing that they also have to be stopped. A new office has appeared, that of the terminal superintendent, and it is my good fortune to be somewhat closely in touch with some of his problems. The old "round house" has become dignified by "locomotive terminal." I believe I am the only railroad man present this morning, and I wish I might be able to add a moiety to these discussions from the standpoint of the railroad man.

2 It should be recognized that the railroads present quite a different problem from that of other industrial organizations. If anything goes wrong with the coal or other conveying installation of a private plant, the finances, or the reputation of the business, may suffer; but with a railroad engaged in public service, we must not forget that whatever happens the trains must operate; that those

who wish to travel expect to go on time; and that shippers expect their goods to be delivered on time, and at the minimum cost of transportation.

3 Railroads are serving the public, in other words; and when making an installation for coaling locomotives they must remember that the installation must not break down. I was much interested in Mr. Baldwin's paper upon this subject, and before speaking of some of the points in connection with what he has given us I want to give an idea of the business proposition that coal conveying and coal storage mechanism for coaling locomotives must meet.

4 I suppose the amount of coal used in American locomotives is valued at about $150 000 000 per annum. There are exactly 55 388 locomotives in operation, which demand a very large amount of coal. The latest type of freight locomotive as used on the Erie Railroad has 16 driving wheels and will haul a loaded freight train two miles long on a level. To take care of the coaling of these locomotives is a large undertaking. Par. 5 of Mr. Peck's paper says: "In a typical plant for supplying locomotives with coal, sand, and water, and for the removal and disposition of the cinders, the coal is carried in an overhead pocket of 1000 tons capacity, or from *one to two days' supply.*" The conveying machinery is in full duplicate, for an hour's stoppage of the large number of locomotives depending upon such a plant would give the terminal superintendent the cold shivers.

5 I disagree with Mr. Baldwin as to the order of importance of requirements for conveyors for railroad use. As in his Par. 76, "freedom from shut down" is the first and most important point; that stands in a class by itself. Number 2 would be speed of coaling; 3, suitable location as regards ground use and locomotive movements; 4, cost per ton for handling.

6 Previous to the present depression in the transportation business, our railroads were often pressed for power to handle the traffic. Under such conditions, a cutting down of the time at locomotive terminals becomes an important consideration. It is equivalent to the purchase of many new locomotives, at a time when new locomotives cannot be purchased because the locomotive building firms may be swamped with orders for a year ahead. I have been connected with a road where I have known one passenger train to be run in 13 sections. This summer I shall be engaged in some locomotive terminal work on a road where some trains ordinarily run in one section or perhaps two, but there may be a sudden change calling for five or more sections of 10 sleepers each. Such congestion may come

when the foreman is cramped by conditions at the roundhouse. Incoming engines must be coaled, fires cleaned, inspected and oiled in the shortest possible time, and the engines quickly turned into service again. A terminal superintendent told me this winter that he had reduced the delay at the locomotive terminal from 2 hours to 20 minutes, without the use of conveying machinery. Of course that is a timed record. Commonly the men do not work at that speed. Labor-saving machinery and appliances at overhead pockets, however, can be handled constantly at record speed, and therein lies one of the advantages of the conveyor, if ground area does not permit pockets with trestle.

7 The location of a coaling plant often presents a more fascinating problem than a game of chess, calling for high skill in operating, in civil engineering, and in motive power departments to avoid interference with main line operation, make good use of the ground, whatever its area and topography, erect suitable structures, prevent interference of locomotives with each other, and aid in inspections and running repairs.

8 Par. 54 of Mr. Peck's paper refers to the cost of maintenance of the trestle type after the first 15 years. Some railroads make use of natural ground contours for the incline, thus avoiding the artificial structure; while occasionally a road, having such an advantageous possibility, chooses to neglect it.

9 Among the many items making up the total cost of coal, I would suggest that too much attention be not given to the cost of fuel to handle the coal. The vast amount of coal used for locomotives, its consequent cheapness, its haulage by the railroad for itself, make the amount used in the operation of the coaling station only a drop in the bucket. Too great attention to this item may divert attention from other more important items.

10 Finally, might I add that sufficient time ought always to be taken in the initial stage of the creation of this or any other engineering property, the stage of "contemplation," which properly precedes the stage of "design." Also, during both of these stages there should be maintained the closest coöperation among the departments of the railroad concerned, especially with the motive power department which is to use the property and will suffer most from its ill design.

Mr. J. McGeorge I had two questions to ask, one of which, upon the best method of conveying cement, has been answered by Professor Carpenter. The second question, which is closely allied with the

first, is that of the best method for getting material like cement into a conveyor. What is the angle at which finely ground cement will flow?

2 I cannot criticise Mr. Titcomb much because of omissions from his paper, for I know how difficult it is to get all relating to a subject in a single paper; but I did notice that he keeps his paper almost entirely to coal and ore handling machinery.

3 My firm is now building an installation of machinery for mixing concrete for some large government locks. We are putting up two crane bridges which take the sand and gravel from barges in the river by means of clam shell buckets, dumping the same into the concrete mixer, from the mixer dropping the concrete into a bucket, and laying it, all by means of those bridges, right in place.

4 There has been no trouble whatever in handling the gravel and sand by means of the grab bucket, etc., but there has been trouble in getting the cement in place to mix with the other ingredients of the concrete. We are handling cement in bulk, through a system of bins and inclined chutes. The bins are built in the form of hoppers and the chutes discharge into a locomotive bin car which runs an average of 600 ft. to the mixer, and the cement is then elevated into a tank and from that discharged into the mixer.

5 Our trouble is a little peculiar. The government engineers insist upon positive measurements. In these positive measurements we had trouble with the cement conveyors. One difficulty was to determine the angle at which cement would flow. I would be glad to hear from anyone that can answer that question.

Mr. William T. Donnelly This discussion has been extended to cover the art of conveying, broadly, but there still remains one branch which has not been touched upon, in spite of the fact that it deals with the oldest known system of conveying —the system that is handling the greatest quantities of material, and one that has been steadily at work since the beginning of the world.

2 I refer to the conveying of solid material by running water. The rivers and streams of the country are every year conveying countless millions of tons of material in their open channels, the force in use being the velocity of the water as it descends from higher to lower elevations.

3 At the present time, man is using the principle exemplified in this method of conveying in the dredging of harbors and rivers, and, to a limited extent, in the conveying of material for the construction

of dams, the velocity being imparted to the water by a centrifugal pump operated by a steam engine, thus bringing in purely mechanical apparatus, and bringing this system of conveying within the scope of the mechanical engineer.

4 It is rather a remarkable fact, that all such plants as have been used for this system of conveying up to the present time have been of a temporary nature; that is, they have been constructed and adapted to operate on a special contract for a limited time, after which they would be displaced or dismantled.

5 The economy or the low cost of handling material by this system is most remarkable. In dredging operations, the cost will range from 10 to 20 cents per cubic yard, with material conveyed a distance of from one-half to more than one mile.

6 From the necessity for conveying large quantities of material is developed the particular kind of conveyor best suited to meet the conditions, and it has seemed desirable to me at this time to call particular attention to this method of conveying solid material by water, as I believe that in the near future there will arise a demand for the continuous conveying of material by this system, and I am convinced that its lack of development has been due to the fact that, up to this time, there has not been a sufficient demand to call for the continuous conveying of material through pipe lines.

MR. HENRY HARRISON SUPLEE This matter of conveying materials by hydraulics has been well carried out. One thing I think is worthy of mention, and that is the handling of ashes on shipboard. On nearly all the ocean liners the ashes from the boiler or from the furnaces are dumped in water and with a steam injector emptied over the side of the ship. The disposition of ashes was one of the most difficult problems on shipboard; it created a great deal of dirt and dust and labor. Now they are forced out with a stream of water and with perfect ease and freedom from dust. I think this device was the invention of one of our members, and is now used universally.

DR. WM. KENT A former president of the New York Steam Company, W. S. Andrews, now deceased, made some experiments, 10 or 15 years ago, on the carrying of anthracite coal in pipes by means of water pressure, and it was claimed that considerable success was reached.

Mr. A. B. Proal (presented by Mr. T. A. Bennett) I speak also for Mr. Proal who is unable to be present and asked me to give some data regarding bridges and towers.

2 There are some unique features in connection with the bridge illustrated in Fig. 14 of Mr. Titcomb's paper. A 36-in. belt conveyor is made use of for handling coal into storage and a five-ton (coal) bucket for taking the material out of storage. This permits putting in coal or other material and taking it out at the same time, giving a bridge of double capacity. In coke plants where two or more kinds of coal are necessary, the field can be divided longitudinally and one kind of coal stored while another kind is being reclaimed. The capacity of the bridge is 700 tons of coal per hour into storage, but this can be increased by increasing the size of the belt. Coal has been taken out of storage by this bridge at the rate of 300 tons per hour. The storage field has a capacity of 750 000 tons and is arranged to be increased to 1 000 000 tons or more if desired. The bridge is equipped with a man-trolley, that is, the man rides with the bucket. Other features are, the load of the bridge on the further shear leg is carried by ball-and-socket bearings, and the bridge is driven through worm-gearing which removes the danger of the bridge running away in a wind storm.

3 The two towers illustrated in the back of Fig. 18 of Mr. Baldwin's paper have established averages, respectively, of 156 and 146 tons per hour in cleaning up 10 000 ton boats. It is important in speaking of tonnage to note that this refers to coal, not ore.

FURTHER DISCUSSION AT DETROIT

THERMAL PROPERTIES OF SUPERHEATED STEAM

By Prof. R. C. H. Heck, Published in May Proceedings

Dr. Henry T. Eddy In considering the comparisons made by Professor Heck of the experimental results arrived at by Professor Thomas, and Knoblauch and Jakob, as referred to by him, the present writer was led to construct the lines expressing the relation of the total superheat in degrees to the total superheat British thermal units at the four pressures investigated by Knoblauch, viz: at 2, 4, 6 and 8 atmospheres nearly, or 28.44, 56.88, 56.88, 85.32, and 113.76 lb. per sq. in. precisely, these lines being suitably interpolated to accord with the results of Thomas as given by him in Fig. 9 of his paper presented to this Society at its meeting in December 1907. The lines so drawn would consequently in general lie between those shown by Thomas in his Fig. 17. It then appeared to me that Professor Thomas had taken some liberties with his original data as shown in his Fig. 9, evidently in order to bring his Fig. 17 into agreement with the idea that saturated steam must begin to be superheated as soon as any heat is imparted to it. That may be the fact and yet the lines on Fig. 17 may be drawn to accord more closely with the data shown in Fig. 9.

2 The object of the present discussion is not to make the slight corrections suggested by these not unwarrantable liberties which Professor Thomas has taken with his data, but to show that his data admit equally well, or in fact better, of a more simple representation which may be regarded as a very close practical approximation to the laws for superheated steam, or steam gas, just as the laws for so-called perfect gases are applied to ordinary gases and are in fact sufficiently close for almost all practical purposes.

3 When we take Thomas' Fig. 9 and attempt to construct his Fig. 17 as he has done, it is found that perfectly straight lines on Fig. 17 represent the mean values of points plotted on Fig. 17 quite as well as the slightly curved lines he has chosen; and had it not been for the supposed necessity for making the lines pass through the origin, no one would have drawn them as they are shown in Fig. 17.

Furthermore, it is found by trial that using fahrenheit degrees and British thermal units as coördinates these lines all pass very nearly through a single point whose coördinates are -34 deg. fahr. and -14.5 B.t.u. from saturation as origin; so that Thomas' experiments may all be very closely represented by the linear equation

$$h = C_p(d + 34) - 14.5$$

in which

h = the superheat in British thermal units.
d = the superheat in degrees fahrenheit.
C_p = the specific heat at the given pressure p.
p = the absolute pressure in pounds per square inch.

4 In order to compare this equation with the experiments of Thomas, let us first compute by it the values of C_p for the data given in Thomas' Fig. 9, and also given in his Table 1 where h and d have been changed from watts and centigrade degrees to British thermal units and degrees fahrenheit by making 1 watt = 3.412 B.t.u., etc.

5 Considering now the values of C_p for these six different pressures there are certain reasons which will appear later for assuming the mean value of C_p for any pressure to be the average of the middle four temperatures. We shall then provisionally assume them to be independent of the degree of superheat as follows: Let these values be plotted as shown in the small circles in Fig. 1. They fall very closely upon a regular curve, except at 165 lb. From this curve then, they may be read as a systematic set of corrected values of C_p for the above pressures or any other intermediate pressures desired.

6 Upon Fig. 1 are plotted, not only the values of p and C_p in Table 2, but also the values of d and C_p given in Table 1, the points for any given pressure being joined by dotted lines to show the deviations of the experimental values of C_p from the assumed mean values, which last are represented by horizontal unbroken lines. These show: First, that the experimental deviations from the mean values adopted follow no general fixed law whatever, such deviations here being represented on an enormously exaggerated scale by reason of the zero of specific heat having been placed so far below the diagram. Second, that the values of C_p for 36 deg. fahr. all lie much nearer together than do the mean lines assumed; being, in fact, about one-half as far apart. Were this drawing together also further indicated by Thomas' experiments, at 18 deg. of superheat, one might be led to think that the superheat lines converge more rapidly as they approach zero of superheat, but the

TABLE 1

p	d	h	C_p	p	d	h	C_p
7	36	19.65	0.4880	115	36	21.5	0.5143
	72	36.2	0.4780		72	41.3	0.5264
	108	54.5	0.4862		108	59.36	0.5202
	144	69.6	0.4724		144	79.6	0.5286
	180	87.6	0.4772		180	97.7	0.5243
	270	130.3	0.4763		270	140.9	0.5112
20	36	20.33	0.4980	165	36	22.23	0.5248
	72	37.5	0.4906		72	42.3	0.5359
	108	55.3	0.4914		108	62.1	0.5394
	144	71.3	0.4820		144	82.23	0.5434
	180	88.7	0.4822		180	101	0.5397
	270	132.9	0.4832		270	143	0.5181
35	36	20.5	0.5000	215	36	22.5	0.5286
	72	38.55	0.5000		72	41.6	0.5292
	108	56.64	0.5100		108	62.1	0.5394
	144	72.9	0.4910		144	81.9	0.5416
	180	91.1	0.4935		180	101.55	0.5421
	270	134.1	0.4888		270	145.1	0.5250
55	36	20.8	0.5043	300	36	22.9	0.5343
	72	39.6	0.5104		72	43	0.5425
	108	58	0.5105		108	63.2	0.5472
	144	75.4	0.5051		144	82.9	0.5472
	180	92.8	0.5014		180	102.7	0.5467
	270	136.5	0.4967		270	149.7	0.5300
75	36	21.15	0.5009	500	36	22.9	0.5343
	72	40.25	0.5165		72	45	0.5613
	108	58.1	0.5113		108	64.5	0.5563
	144	76.4	0.5163		144	83.9	0.5528
	180	95.2	0.5126		180	104.75	0.5572
	270	138.6	0.5026		270	150.8	0.5438

TABLE 2

p	7	20	35	55	75	115	165	215	300	500
C_p	0.478	0.486	0.496	0.507	0.514	0.525	0.54	0.538	0.546	0.557

experiments at 18 deg. do not bear out this interpretation, and so far as they go they confirm the assumption that the lines are straight. As, however, the experiments at 18 deg. were made at a set of pressures differing from the other lines, it is impossible to introduce the data for 18 deg. along with those for other superheats. This evidence will appear later in Fig. 2. Third, that the values of C_p at 270 deg. fahr. are all much lower than the means assumed. Fourth, that were it a fact that the superheat lines have the kind of curvature

shown in Thomas' Fig. 17, the dotted lines in this Fig. 1 should all show unmistakably a form convex upward such as might be indicated by the dotted lines at pressures of 165 215 and 300 lb. Fifth, since corresponding curves derived from Knoblauch's experiments are all very strongly concave upward, and the values indicated by his experiments are for all pressures like the dotted line for 20 lb. in Fig. 1, in which the specific heat decreases at higher superheats, we ought to neglect the doubtful indications of convexity that might be suggested at pressures 165, 215 and 300, especially as the specific heats at 165 are in grave doubt by reason of the anomalous position of some of them so near to or even above those on the 215 line. Sixth, that aside

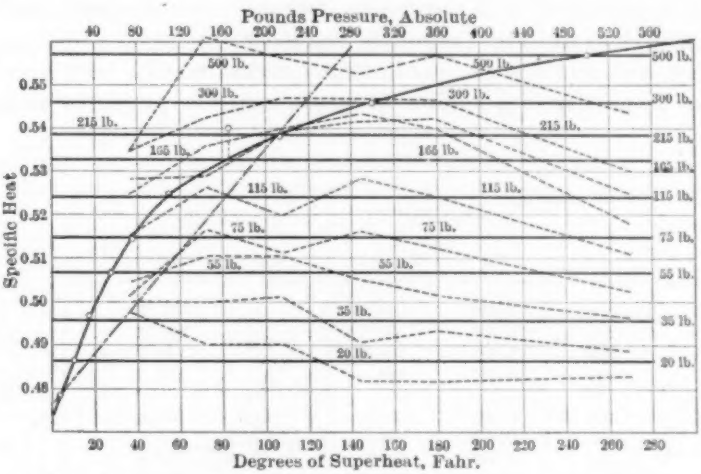


FIG. 1 THE CURVE OF PRESSURE AND SPECIFIC HEAT. THE DEVIATIONS OF THOMAS' EXPERIMENTAL SPECIFIC HEATS FROM THE MEAN VALUES

from these there are no general relations discoverable, and that a constant value of C_p, i.e., a horizontal straight line for each pressure, fits these values of C_p as well as any other kind of curve that could be suggested for them all.

7 Since Professor Thomas has, in his Fig. 17, constructed superheat lines for a certain set of pressures, let us proceed to construct lines for these pressures in accordance with our equation of superheat and Fig. 1. We read from Fig. 1 as follows: In Fig. 2 are drawn superheat lines having these values of C_p as the slope, and all passing through A which may be called the absolute origin of the specific heats of superheated steam regarded as a gas, this origin being situated at -34 deg. fahr. and -14 B.t.u.

TABLE 3

p	0	20	40	60	80	100	150	300	500
C_p	0.474	0.4866	0.4993	0.5087	0.5165	0.522	0.5308	0.546	0.557

8 Having constructed the lines of superheat on Fig. 2, it is possible to construct a superheat curve for any number of degrees of superheat, the coördinates being absolute pressures and British thermal units of superheat. A set of such curves is drawn upon Fig. 2 for the degrees of superheat which Thomas investigated and they show the British thermal units which must be imparted to each pound of steam to superheat it to the degree indicated, on the supposition that our equation and Fig. 1 correctly represent the facts.

9 These two sets of superheat lines and curves may be best conceived of as profiles of a surface in space, wherein pressures and

TABLE 4

		KNOBLAUCH		THOMAS		B.T.U.
p	d	C_p	h	C_p	h	$\triangle h$
28.44	100	0.495	49.5	0.492	51.4	1.9
	200	0.495	99.		100.6	1.6
	300	0.495	148.5		149.8	1.3
	400	0.495	198.		199.	1.
56.88	100	0.50	50.	0.507	53.4	3.4
	200	0.494	98.8		104.1	5.3
	300	0.492	147.6		154.8	7.2
	400	0.492	196.8		205.5	8.7
85.32	100	0.524	52.4	0.518	54.9	2.5
	200	0.51	102.		106.7	4.7
	300	0.505	151.5		158.5	7.
	400	0.504	201.5		210.3	8.8
113.76	100	0.544	54.4	0.525	55.8	1.4
	200	0.522	104.4		108.3	3.9
	300	0.514	154.2		160.8	6.6
	400	0.513	205.2		213.3	8.1

degrees of superheat are used as coördinates in a horizontal plane and the British thermal units taken for the vertical ordinate. The surface has approximately a plane slope with one edge at A, but is warped enough so that vertical planes parallel to the edge A cut the surface in curves not quite horizontal.

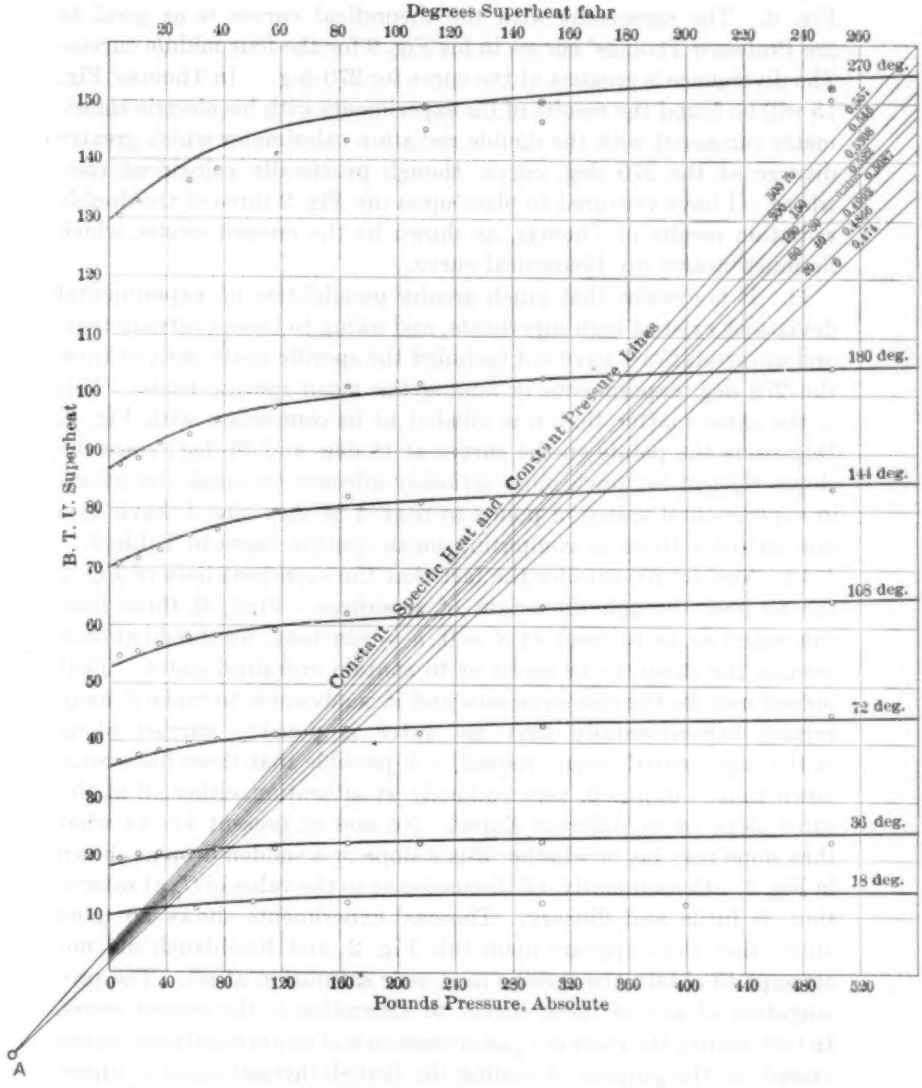


FIG. 2 CURVES OF SUPERHEAT AS PRESSURE AND BRITISH THERMAL UNITS VARY AND OF PRESSURE AND SPECIFIC HEAT AS DEGREES AND BRITISH THERMAL UNITS VARY

10 We have also plotted upon Fig. 2 small circles representing the experimental results of Thomas' experiments in his Table 1 and his Fig. 9. The agreement with the theoretical curves is as good as are Professor Thomas' curves in his Fig. 9 for the four middle curves. The divergence is greatest at the curve for 270 deg. In Thomas' Fig. 15 will be found the results of his experiments with his electric calorimeter compared with the double radiation calorimeter which greatly diverge at the 270 deg. curve, though practically coincident elsewhere. I have ventured to place upon my Fig. 2 three of the double radiation results of Thomas, as shown by the crossed circles, which lie much nearer my theoretical curve.

11 It is evident that much greater possibilities of experimental deviations exist at high superheats, and owing to these contradictions and uncertainties I have not included the specific heats derived from the 270 deg. experiments in finding the mean specific heats. This is the same matter that was alluded to in connection with Fig. 1. Regarding the points on the curves at 18 deg. and 36 deg., since the slopes C_p will be much more seriously affected by small deviations in experimental values of points so near A as they are, I have also not included these in computing mean specific heats in Table 2.

12 Now let us consider the fact that the superheat lines of Fig. 2 fail to pass through the origin of superheat. First, if these lines converged so as to meet at $d = 0$, h = constant, we should at once assume the steam to be moist or to contain entrained water. That indeed may be the case even now and it is advisable to make it more certain experimentally that no water is possibly carried along in the superheated steam; second, it is possible that these lines come down to $d = 0$, $h = 0$, very suddenly at saturation, either all at the same slope or at different slopes. No one at present knows what that slope may be, or whether it is a slope or a sudden drop as shown in Fig. 2. Consequently, all discussion as to the value of C_p at saturation, is futile and illusory. Thomas' experiments shows no more about that than appears upon this Fig. 2, and Knoblauch did not attempt to obtain the specific heat near saturation at all. The prolongation of any of these curves to saturation is the merest guess. In fact, finding the value of C_p at saturation is of no practical importance except for the purpose of finding the British thermal units of superheat h, which is represented with approximate accuracy for technical purposes by our equation or by the superheat lines in Fig. 1.

13 It has been thought by the writer that for professional purposes a chart which would map out the superheat region in a different

manner from the foregoing and which might replace tables of the properties of superheated steam might be most useful. A part of such a chart is given in Fig. 3 based on the foregoing interpretation of Thomas' experiments, and a more complete chart of the properties of superheated steam based on the experiments of Knoblauch is also appended, as Fig. 4, which was constructed in 1907 by Mr. C. F. Shoop under the writer's directions. Mr. Shoop also constructed a plaster model in the spring of 1907 similar to that represented by the

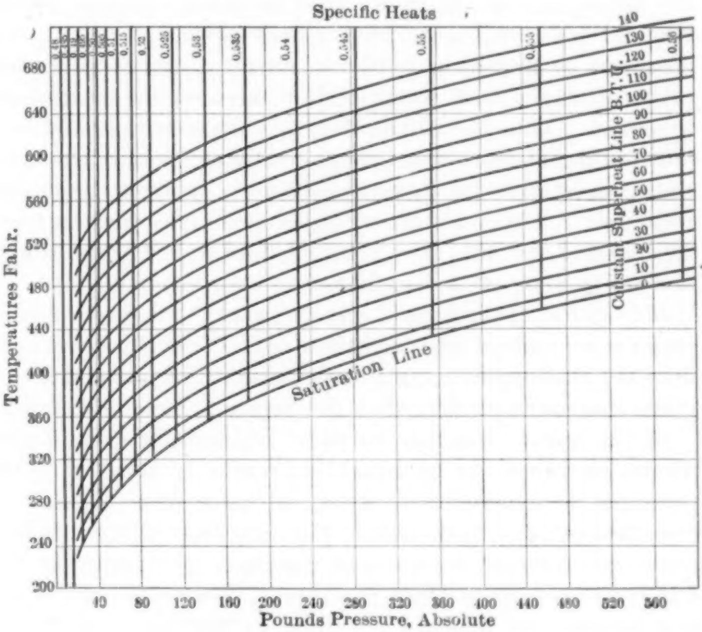


FIG. 3. SPECIFIC HEAT AND BRITISH THERMAL UNITS ABOVE SATURATION OF SUPERHEATED STEAM

Knoblauch curves in Professor Heck's Fig. 4, which may be seen at the University of Minnesota. The curves of constant specific heat in the chart, Fig. 4, are the level contour lines of the plaster model. It is evident that using pressures and temperatures as the coördinates to define the state of the superheated steam, it is possible to represent the numerical value of any property such as specific heat, entropy, etc., as ordinates and so get one surface for specific heats, another for entropy, etc. The level contour lines of such surfaces are the

lines of constant specific heat, constant entropy, etc., upon the chart, from which can be read numerical values of these quantities for any given pressure and temperature. In fact the numerical values of any two properties might thus be used as coördinates and the others to constitute surfaces by using them as ordinates. What I wish to advocate is the advantage in point of simplicity of apprehension of using as primary coördinates, pressure and temperature.

14 The curves of constant specific heat and constant mean specific heat up to any given state may be read upon this chart, and the latter may be depended upon as being accurate according to Knoblauch's experiments, having been integrated and checked with great care by Mr. Shoop, to whom I am greatly indebted for the painstaking accuracy with which he has executed the computations for this chart. Thus the British thermal units at any given state may be found with confidence from this chart so far as Knoblauch's experiments and interpretations justify such confidence. It should be noticed, however, that Knoblauch's actual experiments cover only a part of the area charted, viz: that from pressure 28.44 to 113.76 and from saturation up to about 675 deg. fahr.

15 On Fig. 3 the lines of specific heat C_p being constant for each pressure are vertical lines, and there are no lines of mean specific heat, since the total superheat in British thermal units is h and is to be computed from our equation when the degree of superheat d is known.

16 It seems desirable to show numerically how Knoblauch's results, smoothed out by equalizing curves as they are at the four pressures he experimented upon, compare with Thomas' results, smoothed out as I have done in Fig. 1 by help of lines whose specific heats are constant at constant pressure, by computing values of h in both cases, as in Table 4, where C_p designates the mean specific heat at constant pressure according to Knoblauch, as found upon the chart in Fig. 4. From Table 4 it is evident that within the range of Knoblauch's experiments his results give smaller amounts of superheat in British thermal units for the same degree of superheat than do Thomas' experiments, and the results differ more the greater the degree of superheat. The difference, however, would not be so great if we accept Thomas' results at 270 deg. of superheat as given in his Fig. 9.

17 Table 4 covers the entire range of Knoblauch's experiments. In the part of our chart at higher pressures where values have been exterpolated by Knoblauch from his experiments, his results are also in general smaller than Thomas' by like amounts, except for a

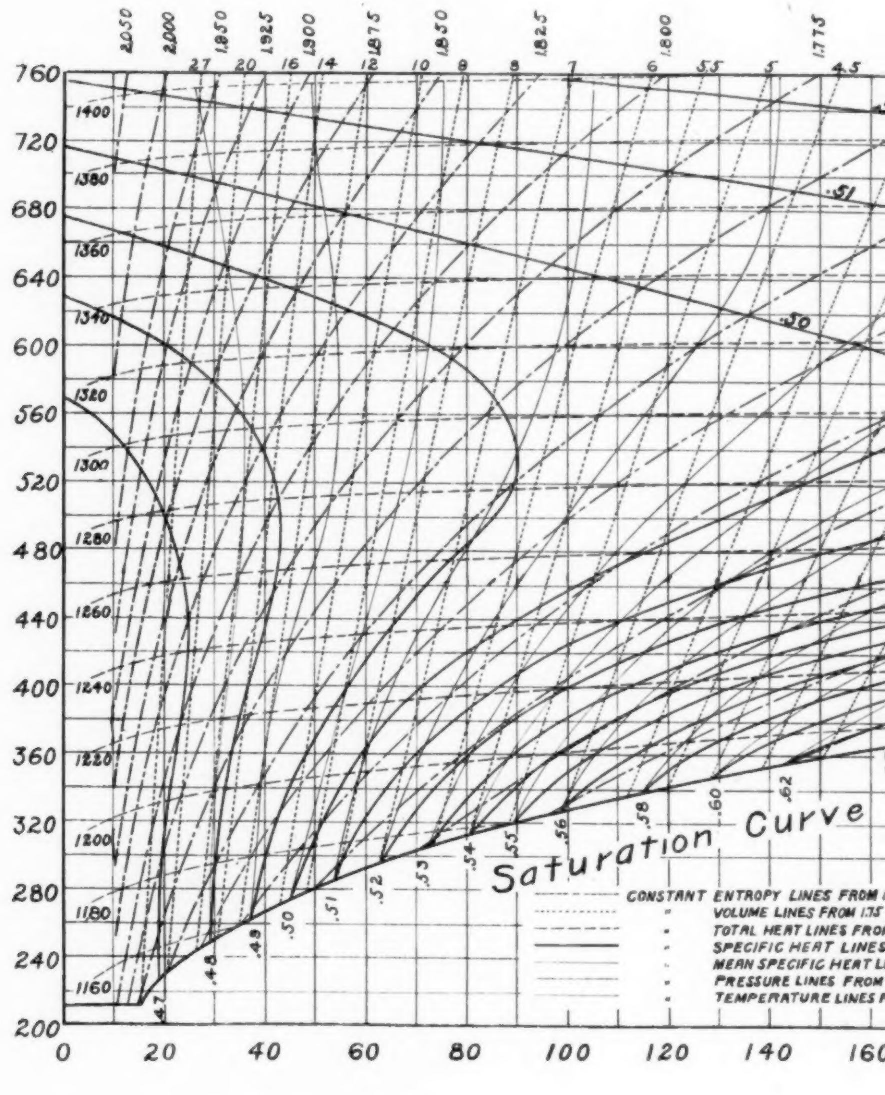
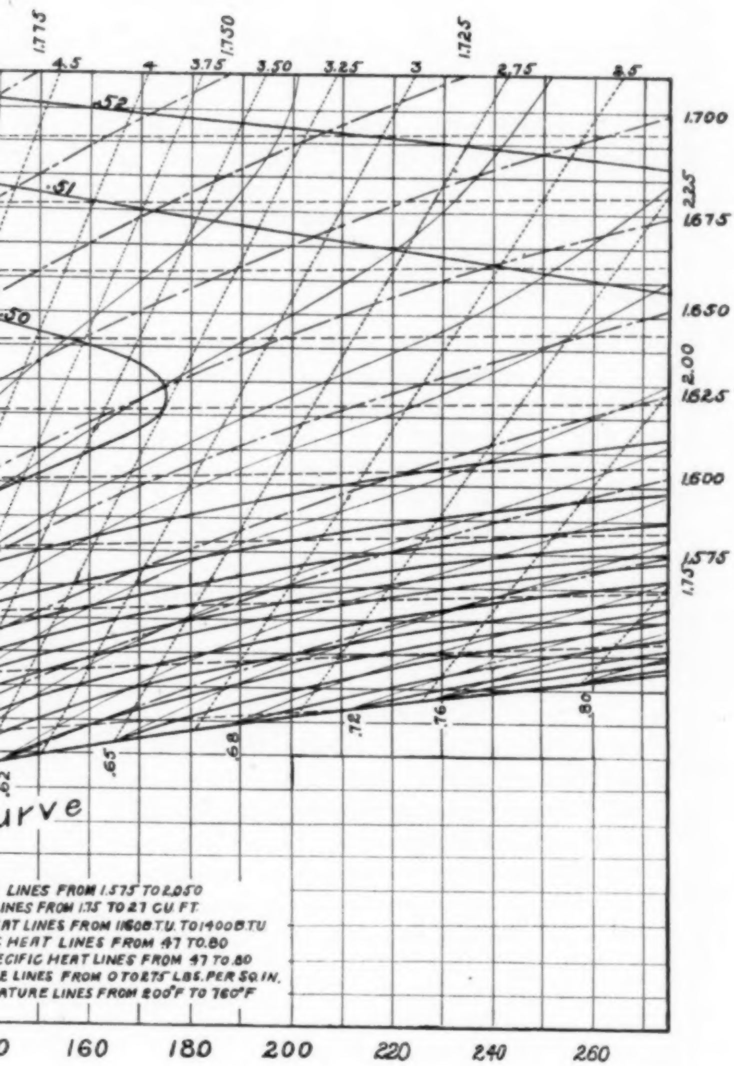


Fig. 4 Properties of Superheated Steam Based on Experi



ON EXPERIMENTS OF KNOBLAUCH AND JAKOB

short space near saturation, where some of Knoblauch's very high specific heats (over 0.6) make his results larger than Thomas' for small superheats under 100.

SUMMARY

18 Professor Thomas' experiments furnish no sufficient basis for the conclusion that his superheat lines are anything other than straight lines.

19 Those straight lines are represented within the limits of experimental error by the equation.

$$h = C_p\ (d + 34) - 14.5$$

20 His experiments, therefore, show that superheated steam may for technical purposes be regarded as a kind of steam gas which has a different constant specific heat for each pressure.

21 His constant specific heats vary with the pressure in a regular manner represented by the curve in Fig. 1.

22 The specific heat lines all start from a single point which may be called the absolute origin of specific heats.

23 The properties of superheated steam may be most conveniently tabulated in a chart in which the state of the steam is defined by its pressure and temperature as coördinates, and numbered curves for constant values of each of the other properties which it is desired to tabulate.

Mr. H. H. Suplee Referring to the question of steam gas, this matter is of importance in connection with recent investigations into the operation of gas turbines, or rather those turbines using gases which have had their temperature reduced by the injection of water or steam. In such mixed turbines the temperatures are so high that the steam is superheated to an extent which causes it to become practically steam-gas, and in the computations relating to such work it is especially desirable to give the correct specific heat of the mixture at the high temperatures of operation.

Prof. William D. Ennis Aside from any question as to the relative weights to be assigned the Thomas and the Knoblauch and Jakob experiments, there is a marked agreement within a certain range, (as Professor Thomas has already indicated), within which it is most assuredly safe to say that the question is "about settled."

The Thomas experiments covered pressures from 7 lb. to 500 lb., abs., with superheat ranging from zero to a variable upper limit (noted on Fig. 6 and 7 of the description, Proceedings, December 1907, pp. 642, 644). This limit was in one series of tests, above 780 deg. fahr., the German experiments were made at pressures of 14.25, 28.5, 57., 85.5 and 114. lb. abs., and at temperatures ranging from a few degrees above saturation up to 400 deg. cent.

2 Considering only the ranges included by both observers, we have the accompanying table, in which no derived values are given, but only those actually obtained by experiment. The italic figures are those of Thomas, obtained by interpolation from his Fig. 6; those in ordinary type are given by Professor Greene, in Proceedings June 1907, p. 1697, from the Knoblauch and Jakob report. From

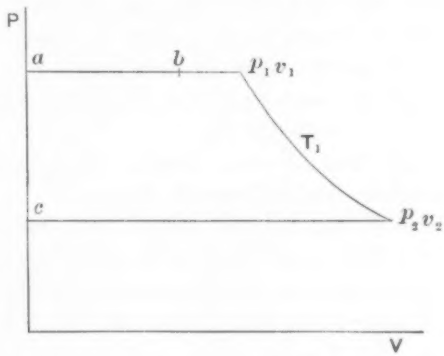


Fig. 1 For Demonstration of Rankine's Theorem

temperatures 392 to 608, inclusive, the difference in the specific heats as independently determined does not exceed 0.01 B.t.u. excepting in the five cases double-bracketed; and in these five cases the maximum disagreement does not exceed 0.013 B.t.u.

3 The heavy black lines indicating the range of what may be called the "close agreement zone" broaden out toward the right, the region of higher pressures, in which Professor Thomas' are the only explorations. The high temperature determinations along the 114 lb. pressure column show a constantly increasing amount of divergence between the two sets of experiments. Equally remarkable is the steady increase in amount of divergence, along each pressure column, as the temperature is decreased. It seems reasonable to assume that the limits of accuracy in experiments of this kind would lead to just such results. At low temperatures, the quantity of heat

measured is small, and more opportunity exists for errors in calibration and correction, while the value of the constant is known to be excessively variable as we approach the condition of saturation; at high temperatures, the radiation losses are greater. In some intermediate region of temperature, the aggregate of the causes of error falls to a minimum.

4 Why, however, should the coincidence between the two determinations broaden out with increasing pressure? The answer is obvious, that if determinations at higher temperatures and under pressures below 114 lb. had been made by Professor Thomas, there is no reason to believe that the heavy black line marking the limits of close convergence would not have been horizontal along the bottom, as it is along the top. In fact, exterpolation of points on the uniform curves by Professor Thomas gave a very nearly horizontal line, dropping only 18 deg. in temperature at 85.5 lb. pressure, and 18 deg. more at 114 lb.

5 Outside the limits of this agreement, we have two courses before us: either the assignment of prior merit to one or other of the determinations, or an interpretative harmonizing such as Professor Heck presents. On a matter which permits of direct experimental determination, it would seem best to base any final tabulation on the experiments. Professor Heck shows clearly the regions in which confirmatory tests are desirable. The searching analysis to which he submits the few results obtained near those regions may corroborate, but it cannot replace, observation.

6 There is possibly a chance of confusion in the use of the "true" and "mean" values of the specific heat. In the expression for entropy,

$$\phi = \int_{T_1}^{T_2} \frac{\delta H}{T} = C_p \log_e \frac{T_2}{T_1}$$

the value of C_p must be the *mean* for a temperature range of T_1 to T_2.

7 In Professor Thomas' Fig. 20 (Proceedings, December 1907) the total heat in superheated steam, calculated from his experiments, is shown not to be independent of the pressure, but to depend on the pressure less directly than does that of saturated steam. At high values, the total heat, while the temperature remains constant, decreases with decrease of pressure for a time, and then increases with decrease of pressure. At low values, the total heat continually increases as the pressure decreases, the temperature remaining con-

stant. Above 1320 B.t.u., there is a definite limited range of pressures throughout which the total heat remains constant at a constant temperature, regardless of the pressure. While the breadth of this pressure zone does not increase as the temperature is increased, the average departure of the total heat from the mean value, at any temperature, while the temperature varies, appears to grow continually less. To this extent, Rankine's theorem that the total heat is independent of the pressure would seem to be confirmed. Rankine's argument rests upon several assumptions, however, mainly with regard to the constants used. The development is as follows: In Fig. 1, ab represents the evaporation of water from a to dry steam at b; $b-p_1v_1$ represents superheating to temperature T_1; $p_1v_1-p_2v_2$ isothermal expansion, p_2v_2-c, condensation at p_2; ca the heating of the water from c to a. If $H_1 =$ the total heat absorbed along cab-p_1v_1; and H_2 the total heat rejected along p_2v_2-c; we have

$$H_1 + cT_1 \log_e \frac{v_2}{v_1} - H_2 = p_1v_1 - p_2v_2 + cT_1 \log_e \frac{v_2}{v_1}$$

and since

$$p_1v_1 = p_2v_2,$$
$$H_1 = H_2$$

That is, the total heat at p_1v_1 is the same as that at p_2v_2; or in other words depends on the temperature only, and not on the pressure. The assumptions are, that superheated steam is a perfect gas, following the law $p_1v_1 = p_2v_2 = cT$, and that the mechanical work done during the heating of the water along ca may be neglected. The latter assumption may be waived; but the law of expansion of superheated steam is at least doubtful. It certainly diverges far from that of a perfect gas when near saturation.

8 The analysis indicates the necessity for an accurate table of the volumes of superheated steam at various pressures and temperatures. Most practical problems involving superheat also involve the use of figures for specific volume. This problem and that of the specific heat, belong together. When both questions are finally settled, I hope Professor Heck will give us a table of entropies, specific heats, total heats, and volumes that can be used along with the saturated steam table and will be at least as accurate.

VALUES OF THE SPECIFIC HEAT OF SUPERHEATED STEAM

Pressure, lb. per sq. in.	14.25	28.5	57	85.5	114
Temperature of saturated steam	210	248	289	316	336
Temperature, deg. fahr.	TRUE SPECIFIC HEATS AT TEMPERATURES STATED				
212	0.463				
230	0.463 *0.502*				
248	0.462 *0.490*	0.480			
266	0.462 *0.483*	0.479 *0.600*			
284	0.462 *0.479*	0.477 *0.523*			
302	0.462 *0.477*	0.476 *0.505*	0.510		
320	0.461 *0.475*	0.475 *0.495*	0.506 *0.568*	0.545	
338	0.461 *0.474*	0.474 *0.490*	0.502 *0.530*	0.536	0.582
356	0.462 *0.473*	0.474 *0.486*	0.498 *0.515*	0.528 *0.553*	0.566 *0.623*
374	0.462 *0.473*	0.473 *0.484*	0.495 *0.508*	0.520 *0.534*	0.552 *0.567*
392	0.462 *0.472*	0.472 *0.482*	0.492 *0.502*	0.513 *0.523*	0.538 *0.545*
410	0.462 *0.472*	0.472 *0.480*	0.489 *0.498*	0.507 *0.517*	0.526 *0.534*
428	0.463 *0.471*	0.472 *0.479*	0.487 *0.495*	0.502 *0.512*	0.517 *0.526*
446	0.464 *0.470*	0.472 *0.477*	0.486 *0.492*	0.497 *0.508*	0.509 *0.521*
464	0.465 *0.469*	0.472 *0.475*	0.485 *0.490*	0.494 *0.503*	0.503 *0.516*
482	0.466 *0.469*	0.473 *0.474*	0.484 *0.487*	0.491 *0.499*	0.499 *0.511*
500			0.483 *0.485*	0.490 *0.497*	0.496 *0.507*
518			0.483 *0.482*	0.489 *0.493*	0.494 *0.503*
536			0.483 *0.481*	0.488 *0.490*	0.493 *0.499*
554			0.484 *0.478*	0.489 *0.487*	0.493 *0.495*
572				0.490 *0.484*	0.493 *0.491*
590				0.491 *0.481*	0.495 *0.489*
608					0.496 *0.485*

Values of the Specific Heat of Superheated Steam—Continued.

Pressure, lb. per sq. in.	14.25	28.5	57	85.5	114
Temperature of saturated steam	210	248	289	316	336
Temperature, deg. fahr.	TRUE SPECIFIC HEATS AT TEMPERATURES STATED				
626					0.499 0.483
644					0.501 0.478
662					0.503 0.477
680					0.506 0.474
698					0.509 0.473
716					0.512 0.471
734					0.515 0.469
752					0.519 0.468

A RATIONAL METHOD OF CHECKING CONICAL PISTONS FOR STRESS

By Prof. Geo. H. Shepard, Published in February Proceedings

Mr. M. Nusim Professor Shepard endeavors to arrive at a rational solution of the problem of the stress distribution in a conical piston without the aid of the theory of elasticity and by means of the principles of statics alone. He considers the equilibrium of an element of the conical shaped piston cut out by two infinitely near meridian planes. In arriving at the equation of equilibrium of such an element, the couple or bending moment acting on any meridian section of the piston is not taken into account.

2 The state of stress in a conical shaped piston subject to fluid pressure can be specified as follows: Consider an element of the piston (see Fig. 1) cut by two infinitely near planes through the axis of the cone and by two infinitely near planes normal to the slant height of the cone (plane *efcd*). That is, an element of the cone cut by a pair of principal planes. The stress on the face *cdef* is due to a couple (or bending moment) and a resultant thrust. (This thrust will not

be normal to the face *cdef* on account of the existence of a shear). Similarly, the state of stress on the face *abcd* can be specified by a couple and a resultant thrust normal to this face. In general both the couple and thrust on this meridian section will vary as we go along the slant height. The resultant thrust, over the whole section made by a plane through the axis of the cone, is of course, given by the product of the fluid pressure and projected area of piston subject to this pressure.

3 If now we consider, as does Professor Shepard, a whole element of the piston cut by two infinitely near meridian planes (see Fig. 2), and desire to find the stress on the face whose trace is *cd*, say, we should take into account besides, the fluid pressure over the element *abcd*.

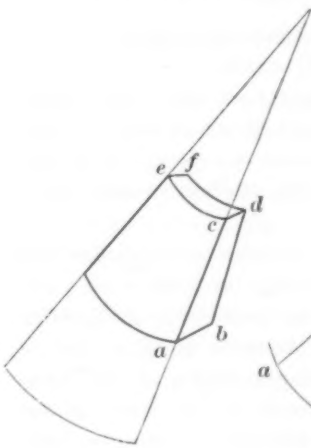


FIG. 1

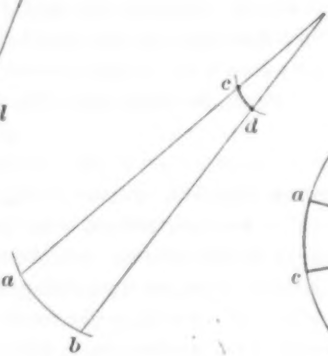


FIG. 2

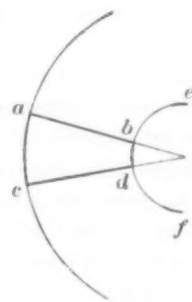


FIG. 3

DIAGRAMS ILLUSTRATING STRESS IN PISTONS

a The resultant thrust on the part of the meridian section between *b* and *d*.

b The resultant couple on the same part of the meridian section; for this couple has a component acting on the face whose trace is *cd*. It is this couple (bending moment) that Professor Shepard leaves entirely out of account in writing down the equation of equilibrium.

4 To bring out this point more clearly, consider a flat piston, Fig. 3. In this case the resultant thrust on any meridian section (or what Professor Shepard calls the resistance to collapse) vanishes, while the resultant couple over the same section has a definite value.

5 Suppose we desire to find the bending moment in the section whose trace is *bd*. It is evident that the couple on the faces whose traces are *cd* and *ab* have a component about *bd* which cannot be neglected. To neglect these couples (on the meridian sections) would mean to consider instead of a solid disc, one that is cut by a series of radial planes up to the circle *ebdf*.

6 The rational solution of the stress distribution in a conical shaped piston cannot be obtained by means of the principals of statics alone. One must have recourse to the mathematical theory of elasticity, otherwise one is apt to arrive at an approximation only.

A JOURNAL FRICTION MEASURING MACHINE

By Henry Hess, Published in January Proceedings

Mr. J. Royden Peirce During the past two years, the writer has had an extensive experience in the practical use of ball bearings under trying circumstances and is glad to see a scientifically constructed machine to determine accurately the friction in such bearings.

2 Engineers in general are not so much interested in the differences in the internal frictional qualities of various ball bearings as they are in their reliability. The internal friction is too small to be an appreciable element in machine construction. On the other hand, an instrument such as the journal friction measuring machine of Mr. Hess should be of great value in detecting the inferiority of many of the ball bearings at present on the market. The final criterion of ball bearing quality is undoubtedly its frictional resistance.

3 If it is true that "Plain, simple words best pierce the ear of grief," it is doubly true that plain, simple bearings are the only ones which stand up satisfactorily under the rough duty of actual usage. It would be very interesting to have a set of tests from this machine of all the different bearings, less with a view to finding the actual friction than to finding the relative advantages or disadvantages of the various ball separating devices under different load and speed conditions.

4 That the friction in ball bearings is extremely small, will be seen from the following personal experience: On 207 ft. of 2 $\frac{15}{16}$ in. line shafting with 32 bearings, a light string passing over a 6 ft. pulley easily turned the entire stretch of shaft with its various pulleys, but without belts.

5 A marble rubbing bed weighing 16 tons and having a moment of inertia of 20 200, continued to run for 25 min. after the power was turned off.

6 By substituting in the well known formula

$$M = \frac{d^2 e}{I\, dt^2}$$

a value of the moment of retardation equal to 56 lb. is found. The inner diameter of the footstep thrust is $5\frac{1}{2}$ in., consequently

$$u = \frac{246}{32\ 000} = 0.0077$$

This includes the slight friction of two radial bearings in the stem and a couple of ordinary bearings on a horizontal quill running light. However, it is an example of regular practical operation and shows what an exceptionally small coefficient of friction ball bearings have. The bearings of these beds were remarkable as having been constructed for a weight of 4 tons instead of 16, and, with one or two exceptions, after eight months' duty they are still running in apparently perfect order.

7 The writer is using bearings on spindles running from 1500 to 2400 r.p.m. operating carborundum wheels. Some of these bear a weight of nearly a ton, which is an excessive load at this high speed, notwithstanding which they last six months with an eight hour day service and then require only a new set of balls. When the power is thrown off, the speed decreases slowly and shows a good coefficient of friction.

8 Improperly designed cages or ball-separating framework may cause excessive wear and heating. An earlier experience showed that by the time overheating called attention to trouble, the bearing was ruined. The clearance and dimensions of the balls and raceways in a well constructed ball bearing are necessarily so exact that any distortion by heating either breaks the balls or ruins the raceways.

9 A testing machine that is sensitive enough to detect small changes in the frictional resistance of ball bearings should render valuable aid and engineers will eagerly await a complete set of experimental data from it.

Prof. John A. Brashear. I would like to add a note as to the early use of ball bearings.

2 In or about the year 1890 we had occasion to take apart and repair an old 13-in. equatorial telescope belonging to the Allegheny Observatory, the mechanism of which was constructed by the Gurleys of Troy in 1856.

3 At the lower end of the polar axis we found a grooved steel ring with about a dozen ⅜-in. hard bronze balls, some of which were in very good condition. I would like to know if any member recalls an earlier date for the use of ball bearings.

Prof. C. H. Benjamin It is evident that Mr. Hess in devising this machine had in mind a machine for measuring the slight friction of ball and roller bearings. Experiments I have made on this class of bearings have shown me that it is very difficult for the ordinary machines to work accurately enough, on account of the very small coefficient of friction with reference to the surface of the journal. I think that the manner of applying the load for the radial measurements, the radial pressure upon the bearing, is not quite clearly shown in the paper. I had some difficulty in understanding just the way in which that load was applied, although it is clearly shown how the amount of friction is to be measured. In designing machines for testing ordinary journals without ball or roller bearings, I think a more simple method should be used.

2 In order to do this several things must be considered. In the first place, the bearing to be tested should itself be insulated as completely as possible from any other bearings in the machine—so that the heat generated by the bearings of the driving shaft or the auxiliary shafts shall not be communicated to the bearing which is to be tested, and this would necessitate a different method of construction from that indicated.

3 According to my experience with journal friction machines, of which I have had quite a little in the last few years, most of the machines are too complicated, and I think there is still room for a simple machine with which commercial journals can be tested under commercial conditions, one capable of carrying the heaviest loads which such journals would be required to bear, and having its test journal insulated as before mentioned. The temperature of the bearing is usually quite as important as the coefficient of friction.

4 With all due credit to the various manufacturers of journal machines now on the market there is still great room for improvement. This criticism, however, does not apply to the machine described in the paper because, as I said before, it is manifestly intended for testing ball bearings and not ordinary journals.

SOME PITOT TUBE STUDIES

By Profs. Gregory and Schoder, Published in May Proceedings

Mr. G. A. Orrok I have been greatly interested in the work done with Pitot tubes by Professors Schoder and Gregory, Mr. White, and the other gentlemen who have been experimenting with them. The curves given by the authors of the paper show in a very good way the eddies and other peculiarities in the flow of the water around bends. In making some power plant tests I have had occasion to measure the condensing water by the use of Pitot tubes placed in the condenser discharge pipe. This pipe was 36 in. in diameter with a number of bends and no piece of straight pipe of sufficient length to free the current of water from eddies. In making the traverses I obtained curves which were almost identical with these curves shown by the authors.

2 Referring to the use of up-stream and down-stream Pitot tubes I have attempted to use them but with negative results, it being almost impossible to secure anything that was consistent; this probably because of the difficulty of keeping the two tubes directly opposite and in line with the flow of the water. I got much better results by using a Pitot tube with a long nozzle in the line of flow, the velocity opening being a surface of revolution, and the static opening four longitudinal slots, 90 deg. apart in the surface of the outer pipe.

3 The constant of this tube I find to be almost invariably, within limits, 100 per cent.

Dr. Sanford A. Moss I have been greatly interested in the portion of this paper referring to use of "down stream readings" of a single opening pitot tube. If it is possible to formulate a definite law between "impact head" (difference between "up stream reading" and "static pressure") and "down stream head" (difference between "static pressure" and pressure shown by tube pointing down stream), the method of using a direct and reversed tube for computation of actual velocity and actual static pressure would be very useful in hydrodynamic work of all kinds. I believe a great deal of experimental work must be done before a definite law can be formulated in the matter. I believe that the relation between "impact head" and "down stream head" may vary with the shape of the impact tube, density and velocity.

2 Some time ago I planned to use up stream and down stream readings for measurement of flow of air in straight pipes for cases

where there were irregularities due to previous elbows, etc., the idea being to obtain true velocity and static pressure by exactly the method used in the paper.

3 Experiments to determine the coefficient were made by use of an impact tube pointing first up stream and then down stream in a jet escaping from an open pipe into the atmosphere. In such a case if the observations are taken at a little distance from the end of the open pipe, the static pressure is known to be atmospheric.

4 The relation between the up stream head and the down stream head varied with the velocity to an extent which was not precisely determined. Most important, however, was the fact that *the down stream head reading was greatly affected by the shape of the tube.* As is well known, the up stream reading, or true impact reading, always gives the velocity head regardless of shape of tube. This certainly does not apply to the down stream reading. A similar experiment was made with a small impact tube in a steam jet of high velocity. The down stream head in this case was much smaller than in other cases.

5 It also seems probable that the down stream reading is greatly affected by slight angularity of the stream lines caused by local eddies, etc. It is certainly true that the static opening or side opening of an ordinary pitot tube is similarly affected. This seems to indicate that a down stream tube, or a side opening for measurement of static pressure, is unsatisfactory, except for parallel flow in a straight pipe. However, in such a case, it seems tolerably certain that the static pressure throughout is the same as the pressure measured by a hole in the pipe wall.

6 The Pitometer Company have developed a system for measurement of flow of water in pipes from sum of down stream and up stream heads. They use an experimentally determined coefficient for multiplying velocity as computed from total head, to obtain the true velocity. This coefficient was originally 0.8, but I am informed by Mr. Cole that 0.84 is shown by recent experiments to be a more accurate value. The reciprocal of this, 1.19, corresponds to the value C in the present paper given as 1.133. There is here a discrepancy of 6 per cent. The values of this ratio C, given in the present paper seem quite discordant and values from 1.09 to 1.21 occur in the region away from the edges of the pipe.

7 In my own work in this matter, instead of using C, the ratio of velocity computed from sum of up stream and down stream heads, to the actual velocity, I have used the direct ratio of down stream

head to up stream or true velocity head. This is $C^2 - 1$. This is the ratio between the directly observed quantities, and different values of it indicate the direct uncertainty in the matter, better than the different values of C which have a much smaller percentage difference. Various values of this direct head ratio are given in the accompanying table.

TABLE OF RATIOS OF DOWN STREAM HEAD TO IMPACT HEAD

(static head − down stream head) ÷ (Impact or up stream head − static head)

Value corresponding to $C = 1.133$ of present paper	0.28
Value corresponding to Pitometer coefficient $1/C = 0.84$	0.42
Value corresponding to Pitometer coefficient $1/C = 0.80$	0.56
Value obtained by writer with air	0.25 to 0.40
Value obtained by writer with high velocity steam	0.28

Prof. H. T. Eddy I fail to see how it is theoretically possible that the readings of a Pitot tube, directed down stream, should have any necessary relationship to the velocity of the stream; or be of any importance. It is theoretically and experimentally known that the readings of a Pitot tube directed up-stream will increase with every increase of stream velocity, however great. But when directed down stream there is a certain velocity of stream which would in any given case produce a suction head as great as the depth of immersion, and any stream velocity greater than that would draw air into the water through the tube, thus making it impossible to read any of these greater velocities by a tube directed down stream.

2 Somewhat the same phenomenon is observed in the wake of a ship, where there is a depression of the water at the stern. No matter what the speed of the ship, the pressure at the stern cannot be less than that of the atmosphere. It would seem to me then, that down-stream readings of the Pitot tube must, in general, from the nature of the case, be illusory. There is a possibility however that for any low velocities a Pitot tube might be so rated experimentally as to make such readings of some value.

COMPARISON OF SCREW THREAD STANDARDS

By Amasa Trowbridge, Published in June Proceedings

Mr. Luther D. Burlingame There seems to be a persistent demand for screw threads of a finer pitch than those derived from the formula for the U. S. or Franklin Institute standard.

2 This demand has been met by the automobile manufacturers by the adoption of a standard having radically finer pitches than the U. S. standard. This new standard, however, does not meet the more important needs of machine tool builders and other manufacturers having similar conditions to deal with, for use as a standard for bolts, screws, studs, etc., as the thread is too fine where used in castings as well as in steel.

3 That the U. S. standard has too coarse a thread to give the best results on the class of work just referred to has been repeatedly pointed out. The Brown & Sharpe Manufacturing Company discarded it 30 years ago and have since used a standard of their own with a finer pitch covering sizes from 1 in. diameter down to $\frac{3}{32}$ in., the pitch of the threads being intermediate between the U. S. standard and the new automobile standard, called the A.L.A.M. standard.

4 In the discussion of the paper by Chas. C. Tyler at the Boston meeting of this Society in 1902, at least five different formulas were suggested for obtaining finer threads. All those taking part in the discussion favored finer threads for the small sizes of screws. Wilfred Lewis said at that time: "The pitch of the standard $\frac{1}{4}$ in. screw is generally admitted to be too coarse and many taps and dies for this size are now made with 24 threads to the inch instead of 20." The pitches recommended by the Committee on Standard Proportions for Machine Screws are a good average of the pitches up to and including $\frac{5}{16}$ in. diameter recommended by the different authorities at the Boston meeting. That these pitches of the A. S. M. E. standard come midway between those in use by the Brown & Sharpe and Pratt & Whitney companies for the small sizes from $\frac{1}{4}$ in. down to $\frac{3}{32}$ in. is a good indication that they give a fair average.

5 The suggestion now made in Mr. Trowbridge's paper that the A. S. M. E. standard for pitches should be generally adopted for these small screws seems in the right direction. He also points out the objection to the U. S. standard as being too coarse for sizes below $\frac{7}{16}$ in. and recommends a standard that is finer.

6 Mr. Chas. T. Porter has consistently and ably urged the adoption of finer threads and gave his reasons in a paper before this Society presented at the New York meeting in 1902 and has continued to urge it since. Arguments in favor of using a finer thread for the regular machine screws up to 1 in. are that the screw is stronger and the tapping cheaper.

7 The British Engineering Standards Committee have, within a few years, adopted a standard for fine threads to provide for the same need that has been pointed out as existing in this country.

8 With Mr. Trowbridge's suggestion as a starting point, it seemed possible that a formula might be devised that would follow the A. S. M. E. standard as far as the $\frac{5}{16}$ in. size and for larger sizes give a new series of pitches intermediate between the U. S. standard and the A. L. A. M. standard, thus, by making one formula cover the whole ground, "killing two birds with one stone," as it were.

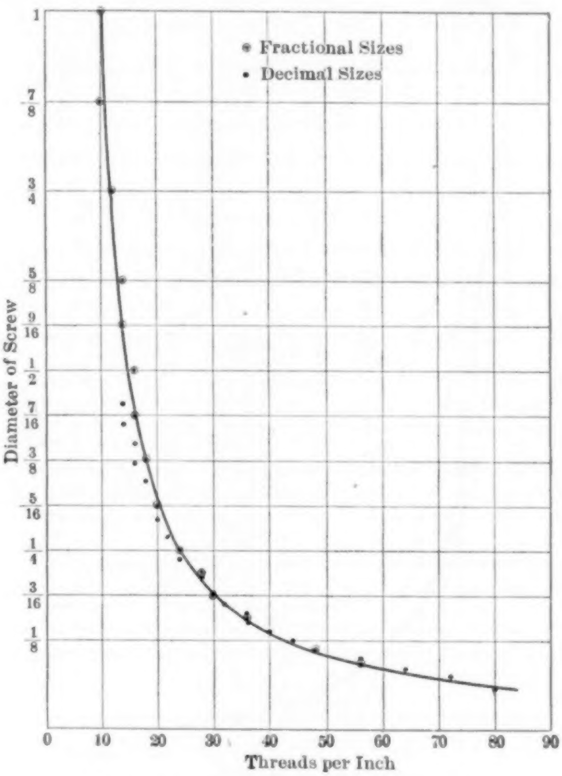


FIG. 1 CURVE PROPOSED BY AUTHOR

9 I have succeeded in deriving such a formula and it produces the curve shown in diagram No. 1, the formula being

$$N = 5.5 + \frac{4.7}{D} \text{ or } P = \frac{D}{5.5\ D + 4.7}$$

When N = the number of threads per inch
and D = the outside diameter of the screw.

10 This curve follows the pitches of the A. S. M. E. standard up to $\frac{5}{16}$ in. diameter as closely as the curve produced by their published formula. That this is possible is due to the fact that the committee departed from their exact formula sufficiently to give even pitches and still further to retain some of the pitches already in use.

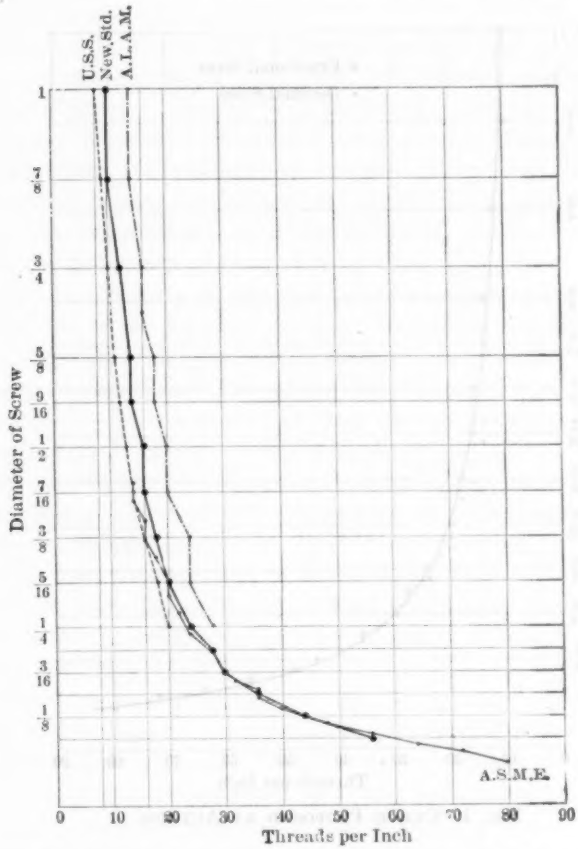


FIG. 2 CURVE FOR PITCHES INTERMEDIATE BETWEEN U. S. AND A. L. A. M. STANDARDS

11 Commencing at $\frac{1}{4}$ in. the new formula gives a series of pitches intermediate between the U. S. standard and the A. L. A. M., as shown in diagram No. 2 and the accompanying table. If it should not be considered objectionable to use odd numbers of threads, the series would follow the formula almost exactly. My preference,

however, would be to depart from the formula to the extent necessary to keep the pitches in even figures, as was done when the A. S. M. E. and A. L. A. M. standards were established.

COMPARATIVE TABLE OF PITCHES

DIAM.	NO. OF THREADS PER INCH							
	U. S. S.	Whitworth Std.	British Fine Std.	A. L. A. M. Std.	B. & S. Std.	Bond Std.	Lewis Std.	Proposed Intermediate Standard.
$2\frac{3}{4}$	4	$3\frac{1}{2}$	6					
$2\frac{1}{2}$	4	4	6					
2	$4\frac{1}{2}$	$4\frac{1}{2}$	7					
$1\frac{1}{2}$	6	6	8					
$1\frac{1}{4}$	7	7	9					
1	8	8	10	14	10	8.5	7.3	10
$\frac{7}{8}$	9	9	11	14	10			10
$\frac{3}{4}$	10	10	12	16	12			12
$\frac{5}{8}$	11	11	14	18	12			14
$\frac{9}{16}$	12	12	16	18	14			14
$\frac{1}{2}$	13	12	16	20	14	14.8	12	16
$\frac{7}{16}$	14	14	18	20	14			16
$\frac{3}{8}$	16	16	20	24	16			18
$\frac{5}{16}$	18	18	22	24	20			20
$\frac{1}{4}$	20	20	25	28	24	25	22	24
$\frac{7}{32}$								28
$\frac{3}{16}$								30
$\frac{5}{32}$								36
$\frac{1}{8}$								44
$\frac{3}{32}$								56

FORMULAE

U. S. Standard

$$P = \frac{\sqrt{16\,D + 10} - 2.909}{16.64} \text{ or}$$

$$P = 24\sqrt{D + 0.625} - 0.175$$

British Fine Standard

Up to 1 inch in diameter

$$P = \frac{\sqrt[3]{D^2}}{10}$$

Above 1 inch in diameter

$$P = \frac{\sqrt[8]{D^5}}{10}$$

Bond Standard

$$P = 0.23 \sqrt{D + 0.625} - 0.175$$

Lewis Standa d

$$P = 0.22 \sqrt{D + 0.25} - 0.11$$

Proposed Intermediate Standard

$$N = 5.5 + \frac{4.7}{D} \text{ or}$$

$$P = \frac{D}{5.5\,D + 4.7}$$

12 The new formula here presented gradually diverges from the A. S. M. E. standard for sizes larger than $\frac{5}{16}$ in., for that standard soon crosses the line of the U. S. standard and becomes even coarser than the U. S. standard beyond $\frac{7}{16}$ in. as seen in Mr. Trowbridge's diagram.

13 The proposed standard has this to commend it, that it is simply an extension of an already established standard rather than the adding of an entirely new one. It is one that would give a proper proportion of diameter to pitch for sizes from the smallest watch screws up to the largest bolts and screws.

14 While on general principles I hesitate to suggest anything which would seem to add to the number of standards for screws, I feel that sooner or later this problem will have to be dealt with and if it is authoritatively done by this Society, it will be settled once for all.

Mr. F. A. Halsey There is no doubt but that there is need of a standard for fine screw threads. Under the present conditions as time goes on people will adopt standards of their own, and eventually there will be great diversity. Under these circumstances it seems wise to take action on this matter and establish a standard that will be generally adopted. Mr. Burlingame has opened the subject, and I move that a committee be appointed to consider the matter. [The motion was seconded and carried.—Editor.]

IDENTIFICATION OF POWER-HOUSE PIPING BY COLORS

By Wm. H. Bryan, Published in June Proceedings

Mr. G. E. Mitchell The engineering profession should feel grateful to Mr. Bryan for placing before this Society a subject worthy of more attention than has been given it in the past. If its importance is fully appreciated it will lead to the adoption at this time of a standard color code.

2 The scheme in general is so well covered by the paper that a few suggestions from the members covering points which may appear to have been omitted or which they know from actual experience tends toward simplicity without loss of efficiency, should help in the adoption of a standard.

3 It seems to the writer that one color should stand for all high pressure steam lines, whether for main engines, auxiliary engines or Holly drip; the same being true also for all exhaust or low-pressure lines.

4 If found advisable to distinguish between pipes leading to the main engines and those of auxiliary engines, in place of using different *shades* of the same color, which after months of service might be hard to pick out, could not the main engines be numbered 1, 2, 3, or A1, A2, A3, etc., and the auxiliary engines 01, 02, 03, or B1, B2, B3, etc., and these designations be placed at the point of an arrow marked on the pipe showing direction of flow in the pipe. This would reduce the number of colors.

Pipe Marked for Superheated Steam

Pipe Marked for Saturated Steam

SYMBOLS FOR IDENTIFYING PIPE LINES

5 I believe also some distinguishing mark could be used to separate *saturated* and *superheated* lines and still use the same body color showing high pressure steam. Assume, for example, that red is adopted for a high-pressure pipe line. If this pipe line was for superheated steam, it might be indicated by a black circle containing a white letter "S" placed at the point of the arrow indicating direction of flow. Another pipe line for high pressure saturated steam would also be painted red, but at the points of the arrows there would simply be a number indicating the engine or auxiliary to which it was connected, as mentioned above. (See Fig. 1.)

6 This scheme requires but one color for all high pressure steam lines, and when possible I believe the darker colors should be used in the boiler room, as the lighter ones will become easily discolored, owing to the nature of the work around a battery of boilers.

7 A standard color chart hung in the chief engineer's office or under the licenses of engineers and firemen would prove a ready reference.

8 There is one other point I believe should be considered at this time as it bears a close relation to all power house piping and that is valves. Some valves open right-handed and some left-handed, and while the writer is firmly of the opinion that only *rising* spindle valves should be used in important work, there are many places where such is not the case, and even though the valve manufacturer is today marking in some way the "direction of closure," are they marked so prominently that in emergency cases, where seconds often count, the operator can feel sure of this direction? Can we not mark valves so that the direction of closing cannot be mistaken?

Mr. Fred'k W. Salmon It appears to me to be beyond question that we need and should have at the earliest possible date a standard system of marking piping in power plants both by colors and by certain conventional lines on drawings and plans.

2 I think such a system should be uniform throughout this country if not throughout the world, and in government work as well as private work, and further it should include simple markings for the different kinds of valves and cocks, such as single gate, double gate and globe valves, plug cocks, etc.

3 I would strongly advocate the use of conventional lines and marks which may be put directly on the tracings of a power plant, steamship, distillery or whatever the drawing may apply to, and so appear clearly on every blue print or copy issued from it; and to further its general use the system should be very simple and the lines or marks easily made with the ordinary drawing instruments.

4 It would be advantageous to have small pieces of thin tracing cloth or paper, about 2 in. by 3 in., printed in black ink, with heavy faced type, stating that the A. S. M. E. standard piping marks are used on that drawing, and giving below the conventional marking and lines, and also stating the corresponding color. One of these labels could be secured by transparent varnish to each tracing on which piping appeared, so the draftsman working upon the drawing would always have the table before him, and a correct copy of the table would appear on every print made from it.

5 If the insurance companies were brought to realize the value of such safeguards to property and life as this, it is possible they would arrange a schedule of rebates to be applied to risks where such useful

schemes were in use, and these rebates in turn would influence the adoption of safeguards.

Mr. John W. Lieb, Jr. This paper is of a character not requiring extended discussion as it speaks for itself.

2 The need of some conventional method of indicating the uses to which the different pipe lines are put in power plants is obvious. The importance of continuity of service in the conduct of power plants is paramount, and therefore the necessity of meeting promptly any emergency that may arise is evident. It is undoubtedly true that such a system as indicated in the paper would go far toward achieving such a result. It is evident, however, that because of the different kinds of service it has been necessary to cover, it would be very difficult to arrive at a sufficiently comprehensive scheme without exhausting all the prismatic colors. Stencils also should therefore be used, indicating by lettering the detailed use of the pipes; but this should not prevent the adoption of a general color scheme.

3 This is a matter which might well be made the subject of standardization, and the only question is whether the matter has been sufficiently considered in all its phases to be ripe for standardization at the present time.

4 It would appear to me that following the suggestion made by the writer of the paper, this matter might well be laid before the Council of the Society for their consideration before action is taken leading to the appointment of a committee; and I would therefore move that the suggestions made in this paper as to the formulation of a code of symbols for power house piping be referred to the Council of the Society for consideration. [This motion was seconded by Mr. L. D. Burlingame—Editor.]

Mr. George A. Mattsson This coloring of pipes has been done on every steamship on the Great Lakes. Take for instance the new D. & C. steamer, City of Cleveland, completed a few months ago. The pipes are all colored, and we not only colored the hot and cold water pipes, the air pipes, suction and discharge pipes, etc., but also the conduits for electric wires. The pipes on such a ship are so numerous that the coloring is necessary.

ECONOMY TESTS OF HIGH SPEED ENGINES

By Messrs. Dean and Wood, Published in June Proceedings

Mr. Geo. H. Barrus The striking thing which these tests show, to my mind, is that unpacked piston valves, or balanced valves which do not automatically take up wear, often leak so much in ordinary service that engines equipped with them are most wasteful steam users. I think that engineers generally who have had occasion to test such engines fully realize this fact. It certainly conforms to my experience in the matter. Beyond this, I see nothing especially striking in the results, and I am unable to draw any positive conclusions from them as to the comparative economy of the different types of valves.

2 In the case of balanced flat valves with distance strips, leakage may be due to improper adjustment, to wear, or to both of these conditions, and two engines of the same type, having had the same period of service and the same care, might show quite different economy owing to the difference which might exist in the adjustment of the distance strips. Consequently, the No. 4 engine referred to in the paper, with four flat valves, cannot rightfully be compared with No. 1 and 2 engines having single flat valves as is done in the author's comments, unless it is known positively that the adjustments of these strips in the different cases were as favorable in one as in the other. Upon this point no information seems to have been obtained, and, in its absence, any comparison appears to me useless. The same may be said as regards a comparison of flat valves and piston valves. The former may have been in proper adjustment, or they may never have been in proper adjustment, apart from any question of the effect of wear or other points of design.

3 It would have added much to the interest of the paper if there had been included in the list of engines one or more having single piston valves with ring packing, in which the wear of the valve is automatically provided for by means of the flexible ring. There are many engines of this kind in successful operation, and they form one of the representative classes of high speed engines. I have tested many of these packed piston valves by putting full steam pressure on them with an open vent and making the customary ocular demonstration of leakage, both with new engines and with those which had been in practical service for various periods up to one year or more, and I do not recall a single instance where leakage of any consequence revealed itself.

Mr. C. A. Dawley The tests reported by Messrs. Dean and Wood are of much interest to steam users when considered as a collection of isolated tests on automatic engines under various conditions of type, size, speed, load, steam pressure, steam quality, back pressure, over-rating and under-rating. They should be of value as an indication of what results in the use of steam are being obtained in the operation of these individual engines. However, it is a principle too well recognized to be overlooked, that in seeking for reasons and arriving at conclusions as to the cause of variations in performance, it is necessary to change conditions one at a time, if their relative effects are to be properly estimated. For this reason it seems to the writer that any sweeping dictum as to what type of engines manufacturers should or should not build, is not warranted as a conclusion from these seven engine tests. It seems especially rash to conclude that reduction of clearance space, shortening of ports and separation of admission and exhaust passages, accomplish nothing, merely because the two engines in these tests, which were so arrranged, were fitted with pressure plate balanced valves which leaked.

2 It is well understood that a type of valve which follows up its own wear is always tighter than one that depends on exact fit for tightness, and that as both types of valve wear with use, the former maintains its original economy much better than the latter. The fact that the four-valve engines tested were fitted with leaky valves should not be used as an argument against four-valve engines.

3 The rather poor results obtained in most of these tests are attributed by the authors chiefly to valve leakage. There is another factor not touched upon, which may be equally important; viz: piston leakage. This is usually assumed to be small, chiefly because (on account of condensation and other uncertain elements) there is no reliable way of determining its extent. The writer has had occasion to make numerous tests on piston leakage in air-compressor cylinders where all other conditions except the piston and its packing were kept constant and where all other sources of loss were measured with great care. Measurements made on the same machine (of a size in the same range as engines reported in these tests) showed leakages, expressed as a percentage of piston displacement, of 4.39, 7.75, 13.30 and 14.85, or approximately 5 to 15 per cent.

4 These results are under rated conditions of load, and the worst of them was obtained with a type of piston rings in very wide use in both steam and air cylinders. Smaller machines show worse and larger ones, better results.

5 It cannot be said that air and steam have equal inherent "leakiness" and I have had reason to believe that there is a difference but these results would certainly hold true relatively, if not absolutely, for steam as for air. I merely wish to bring out the necessity of considering a factor usually neglected in investigations of causes for lack of expected economy in steam engines.

MR. RICHARD H. RICE The tests reported by Messrs. Dean and Wood are extremely interesting, as showing the actual performance of engines in service in the power stations of users, and receiving the ordinary care which such power stations provide. Tests of this nature are very difficult to obtain, because they involve tests of the boiler plant as well as of the engines, and considerable changes must be made in the arrangement of the station to permit these tests to be made with accuracy. There is also difficulty because of the commercial requirements of the customer which always interfere to a great, and usually to a prohibitive extent. The Society is therefore fortunate in obtaining such accurate and satisfactory tests on these non-condensing machines.

2 The tests indicate conclusively that the refinements of construction which have been considered necessary to secure high economy in high-speed non-condensing engines, such as small clearance, prompt valve action, accurate adjustment of the compression, etc., are of no importance whatever when the engine has once been put into service, since the question of leakage is shown to be the all-important element in the efficiency of an engine of this type. All engines containing valves sliding on their seats, whether these valves are piston, plain slide or balanced valves, one, two or four per cylinder, are shown to be subject to this great source of loss, and therefore, until this difficulty is eliminated, consideration of the minor refinements is unnecessary.

3 The tests also indicate one of the chief reasons for the superior economy of the compound engine, as compared with the simple engine, which is, that leakage through the high-pressure cylinder is all caught in the low-pressure cylinder and there utilized. In the low-pressure cylinder, again, the leakage is less in amount and importance, due to the low pressure involved. This point has undoubtedly much to do with the success of this type of engine. Unfortunately, however, the compound engine is not well adapted for non-condensing use, since it requires to be loaded at about the point for which it was designed, otherwise the distribution of work in the two cylinders becomes deranged, and for light loads the low-pressure cylinder actually increases

the water rate of the engine, as compared with a simple non-condensing machine, due to the formation of loops in the indicator cards and consequent development of negative work in both cylinders. This fact has been proven time and again in practice and on varying loads—for instance, in railway work where the load is particularly variable several power stations have been entirely remodeled, due to the great cost of operating them with non-condensing compound engines.

4 One significant feature of the results of tests reported is shown by plotting water rate curves of the engines, by which it will be seen that the proportioning of the engine cylinders is in many cases entirely wrong for the generators to which they are attached, and there is tremendous variation in the adaptability of the engines to their generators, which fact seriously affects the showing made by various engines. For instance, engines A, B and C show marked improvements in efficiency by increase of load, and it is quite evident that if the load were increased to a considerable extent on these machines the efficiencies would be increased correspondingly.

5 The tests are too few in number to give any reliable information as to the relative freedom from wear of any given type of valve. I regard it as entirely accidental that the piston valve engines, A and B, give better results than any of the other types. It is however disappointing, as pointed out by the authors, that the four-valve engines which have been provided with considerable additional complicated mechanism for the sake of securing higher efficiency, should produce results no better than the other types; and it is quite evident from these tests that additional complication introduced into valve mechanisms for the purpose of securing greater efficiency is complication thrown away, until a type of valve is used which will be sure to be tight.

6 Some years ago my attention was called to the double poppet valve, which is so widely used in Europe for engines of slow and moderate speeds, and which at that time appealed to me as being the only valve which could be properly used with highly superheated steam. Experience with this valve was entirely satisfactory under all conditions, and if such a valve is properly made, with its seat so designed as to expand at the same rate and under the influence of the same temperatures as the valve itself, I believe that this valve is the best one for all classes of engines where the speed of the engine is sufficiently low to permit the valve to seat without noise or shock, eliminating as it does all rubbing surfaces from the valve itself. Experi-

ence with *single* poppet valves in steam turbines has indicated that such valves, if once tight, will remain tight indefinitely; that wear simply brightens up the seats and puts them in better condition; and the double poppet valve, if so designed as to be free from differential expansion with its seat, is also likely to maintain its tightness.

7 Comparing the engines tested with similar capacities of steam turbines, the water rates obtained in these tests are considerably higher than those which are obtained from the turbines. It may be objected that the turbine water rates are obtained from machines comparatively new, and that it is not known that turbines will retain their original water rate indefinitely. This contention is however not in accordance with the facts.

8 A number of tests have been made on turbines which have been in constant operation for from one to two years, and these tests show, within the limits of accuracy of the test methods that the water rates of the turbines tested are the same after the completion of the above mentioned periods of use as at the beginning. Unfortunately, the conditions under which these tests were made have not been such as to permit them to be properly laid before the Society at this meeting, but it is expected that in the near future a number of tests of the same high class as those under discussion, can be made by independent engineers, which will show the actual performance of turbines of similar capacities after extended periods of use, for comparison with the figures given in the paper under discussion.

9 Comparing the engine with the turbine, we have in the engine a number of rubbing surfaces, deterioration of which immediately affects the efficiency of the engine to a serious degree. These rubbing surfaces are contained in the valves and seats, and in the piston packings. It is true that these parts are lubricated by well known means, but this lubrication is essentially imperfect, since it depends upon the atomization of the oil in a current of steam, and its deposition on the cool surfaces of the valves and cylinders, a process which is subject to great variation in effectiveness. This is particularly so where the steam carries considerable moisture, in which case it is well known that the lubricant becomes rapidly washed off from the surfaces, instead of being left on them in a position to do its proper work. What occurs in the case of considerable moisture is only a well-defined indication of what actually takes place to a lesser extent at all times during the operation of the machine and this irregular and uncertain application of the lubricant occurring continually during the operation of the machine, results in the wear which has caused

the leakages which have in turn caused the loss of efficiency noted in the engines under discussion.

10 In the steam turbine we have no parts exposed to the steam and affecting the efficient operation of the machine, which require lubrication; and discussing now particularly the Curtis turbine, with which I am most familiar, we have no moving parts affecting steam consumption which come in contact with other parts, either moving or stationary, with the exception of the admission valves, which are of the single-poppet type and which wide experience has shown remain tight when once put in this condition. After the steam has once passed into the nozzles, the only parts which can affect steam consumption by deterioration are the nozzles and the revolving and stationary buckets.

11 No cases have come to my notice in a wide experience covering many hundreds of turbines in operation for periods of from five years downwards, where nozzles have deteriorated with the exception of one case, where by error some improper material was introduced into the nozzles of one stage of a Curtis turbine exposed to the influence of salt water in the steam. In this case the nozzles rapidly deteriorated, but other nozzles constructed of the proper material in the other stages of the same turbine, showed no deterioration whatever.

12 The revolving and stationary buckets, formerly made of steel, in some cases showed deterioration after two or three years' use in cases where salt and moisture were present in the steam, and where the turbine was frequently allowed to shut down and allowed to stand for a sufficient length of time to allow rust to form on the surfaces. Present practice with this particular type of turbine is to use bronze buckets, which are not subject to any deterioration, so far as experience has demonstrated. It is true that damage has occurred to both nozzles and buckets by the passage through the turbine of foreign substances like bolts, nuts, core sand, and other improper material, but such damage has always occurred during the early life of the turbine when the valves are opened and the steam acquires sufficient velocity to pick up such foreign substances and carry them along through the moving parts. Engines are not exempt from this source of damage, as has been repeatedly demonstrated to me in my connection with engine construction.

13 The question of packings needs some discussion, since it might be claimed that in the turbine there are packings which could cause leakage, and this leakage would very likely cause loss of efficiency. Experience shows, however, that the packings of a turbine are fully

as reliable and remain fully as tight as those used in ordinary engine practice.

14 Non-condensing, simple, high-speed reciprocating engines in sizes up to 150 kw. capacity are therefore subject to the following causes of loss of efficiency:

a. Wear of valves and pistons, causing excessive steam leakage.

b Loss of efficiency on variable load, due to the disturbance of cylinder conditions, which is well known to occur under such conditions of loading.

15 Compound non-condensing high speed engines with steady loads at about the rated capacity will eliminate the first source of loss mentioned, but are unsuited for varying loads and are much more expensive.

16 Steam turbines, while showing a lower steam efficiency under test conditions, maintain their efficiency indefinitely and also maintain an average efficiency under varying loads equal to the actual test efficiency at the load which is the mean of such variations. The efficiency of such turbines is at all times considerably better than the engines tested by the authors of this paper.

17 Therefore, for all practical conditions, the non-condensing steam turbine of capacities up to 150 kw. is more effectual, from the standpoint of maintained efficiency, than any form of high-speed reciprocating engine.

Mr. R. C. Stevens The authors of this paper have performed a good service in bringing out so prominently the peculiar advantages possessed by some types of valves, which are initially steam tight and which by design and construction compensate for wear and remain tight instead of wearing loose. Valves which are non-adjustable for wear of necessity have two or three thousandths of an inch clearance when new, even if fitted with the highest mechanical skill, and after operation for short periods permit the leakage of much steam into the exhaust without useful work being accomplished thereby.

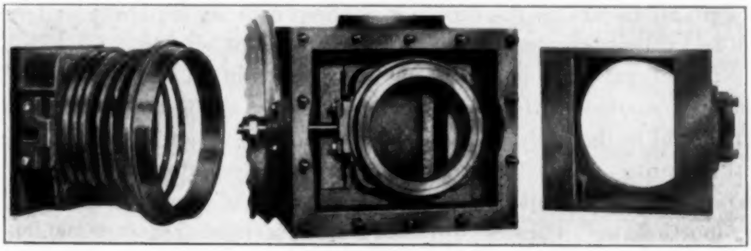
2 The writer shows herewith a valve of the adjustable type not included in the reported tests of Messrs. Dean and Wood, but now in use on some 8000 engines. It has so proven itself that the makers guarantee it will remain tight for a period of five years without adjustment or attention.

3 The valve is accurately fitted to its seat by scraping and 80 per cent of its area is relieved of pressure by a balance ring which rides

against the steam chest cover (not shown). This ring is free to revolve and changes position with every stroke of the valve, preventing any creasing or cutting of valve seat, ring or cover. Steam packing rings prevent leakage between the ring and hub of valve. The construction is such as to allow large port area, which is necessary to proper steam distribution and the development of the full cylinder power. Twenty per cent of steam pressure holds valve in steam tight contact with the seat and takes up all natural wear. The valve is free to lift from the seat if water enters the cylinder.

4 The writer has recently examined two engines, one 13 years old and the other 17 years old, and the valves and seats are in perfect condition—steam tight without having been refitted in that period.

5 An acceptance test was recently made on a 375 h.p. 16 in. and 27 in. by 18 in. tandem-compound engine fitted with this valve, which



VIEWS OF BALANCE RING SLIDE VALVE

at 227 r.p.m., 129 lb. initial pressure and 1 lb. back pressure, developed an indicated horse power on 21.1 lb. of steam per hour. This engine had been in use nine months, but the makers were so confident that the valves would be in good condition that the steam chest covers were not removed, and the engine was tested without any previous adjustment. The writer believes that this is the lowest recorded steam consumption for an engine of this size and type, operating under these conditions.

6 In a plant in Buffalo, where there are five engines with this type of valve, which have been in service two years, power is being developed for less than one cent per kilowatt hour, all charges considered.

7 The steam tight valve has demonstrated its efficiency and the wonder is that, in these days, when consulting engineers so often call for steam economy guarantees, they do not also insist on guarantees

of maintenance of economy—not what the engine does when new, but what it will do when old.

Mr. John C. Parker I should like to make a few remarks as to the significance of the given data for the user of engines and for the consulting engineer who may have to do with small plants of the type considered.

2 I judge from the paper that the type of plant referred to is that commonly found in office buildings and light manufacturing plants. It has been my privilege, working in conjunction with Mr. H. W. Peck, to investigate a large number of such plants, and from what we have seen of the operating conditions ordinarily obtaining, I believe one explanation of the failure of small engines to maintain good efficiency to lie in the inferior oversight provided. We have found the stokers attending between times to the engines, and discharging all sorts of other functions about the building. These men are generally underpaid and usually render a rather under-intelligent service.

3 In general there is a very low load factor in an office building, most of it carried in the winter time when the days are dark and stormy, and in the early and late part of the day—I refer especially to lighting plants. The consequence is that if an adequate sum of money is expended in plant operation the cost per unit of power delivered is disproportionate. This is obviously a determining factor which militates against the use of more expensive, although perhaps more economical, generating units. Such a condition seems to be inevitable and perhaps justifies what seems to be niggardly maintenance.

4 It is a question whether the suggestion made by the writers in regard to the use of really high class engines would result in greater economy. It seems to me it would rather tend to enhance the difficulties mentioned, since such engines require adequate operating attention. It is largely a matter of indifference during the months when heating is carried on whether the engine is of high or low efficiency type. In most office buildings and in many industrial plants a large amount of steam is required for heating in proportion to the amount of power required of the engine, so that questions of efficiency are of less importance than questions of investment. Most of the heating, however, is done during the four winter months; in the summer months a particularly difficult situation arises in that the efficiency of both engine and boiler in a plant of this type necessarily becomes bad, because of the greatly reduced load and load factor. We have found it eminently desirable in such a plant to improve the

efficiency of the boiler plant during such periods of reduced load, by bricking over the bars to reduce the grate area.

5 A more extended discussion by the authors of the paper as to the plant efficiency obtaining in these smaller plants, taking the plant as an integer, including boilers as well as engines, would have been as worth while perhaps as the discussion of engine efficiency alone.

6 Another point in the paper which I believe worthy of still further study is the depreciation shown by these reports. I think none of the engines had been in commission over four years, some less, yet many of the engines already show considerable deterioration. It would be interesting to know just how long the life of engines of this type might be taken to be.

7 One speaker has referred to the excellent characteristics of the steam turbine in the matter of sustained efficiency. While in general the steam turbine might be expected to sustain its efficiency fairly well in smaller sizes this sustained efficiency may be purchased at the price of low efficiency throughout. In one case within our experience a manufacturer using exhaust steam to heat his plant during the winter, who had difficulty in securing a sufficient supply with his old reciprocating engine, had no difficulty whatever after the installation of a steam turbine. While such a statement is not scientifically exact and is subject to some question it is indicative of much. Indeed we should expect to find the splendid qualities of the larger steam turbine absent in the smaller sizes, because of the necessarily lower peripheral velocity and the impossibility of securing the same clearances.

8 It seems to the speaker that there may be a moral to the story of large steam consumption brought out by this paper. While I do not wish to offer a panacea for dynamic ills and while my commercial associations may bias my judgment it seems to me that a great deal of the class of work done by these small engines can be done advantageously by the modern processes of coöperation and consolidation. If all the small power users in a community can pool their interests, put in a small central power-generating and heating plant and distribute on a basis of equitable costs from this, they may derive the benefits ordinarily resulting from the use of large, high efficiency steam turbines with skilled attendance. It is a matter of indifference whether this pooling be done by a coöperative corporation or by an outside concern. If an electric generating company whose plant operation is properly carried out, which does not demand excessive profits and whose securities are not over-inflated, can be found—and the speaker believes that this is by far the rule rather than the exception—then

the proper procedure would be to derive the power supply from such central station.

Mr. F. A. Halsey The reference by Mr. Rice to the tendency of double-beat poppet valves to leak because of the differential expansion of the valves and seats, leading to a slight opening at one or the other seat according as the valve or the seat casting expands the more, leads me to call attention to a construction of this form of valve used by the Nordberg Manufacturing Company, whereby this defect is overcome. In the usual construction the seat faces are parallel with one another, and it is from this that the defect arises. In Mr. Nordberg's construction the two seat faces are made to radiate from a common center, with the result that any slight difference of expansion leads only to a sliding of the surfaces on one another, and not to their separation, as in the usual construction.

2 The accompanying Fig. 1 and 2 illustrate two valves of this construction, one being a steam and the other an exhaust valve. In transmitting the blue prints from which these illustrations are made, Mr. Nordberg makes the following observations:

3 My experience is that the only steam distributing organ in an engine that can be depended upon to be permanently tight, if it is so originally, is the double-beat poppet valve of the type shown by the enclosed blue prints, which are the valves used in the high-pressure cylinder of the Champion compressor for steam and exhaust. If such valves are not tight originally, they will never get tight by working.

4 We built a poppet valve engine in the year 1892. This engine is 24 in. in diameter by 48 in. stroke and the valves of this engine have, since the time it was started, never been taken out and are known to be tight today. In this particular case we have considerable wet steam, which, as a rule, is detrimental to a poppet valve, as also is any dirt or sediment that may come from the boiler.

5 The material of which a poppet valve and its seat are made is a very important point if the valve is to be depended upon to keep tight. Certain materials and bronze mixtures for some reason are absolutely unsuitable for valves of this kind, and if such material is used my experience is that it is impossible to make the valves tight.

6 Regarding the matter of expansion, my practice has always been to design such valves in such a manner that the surfaces of the seats, if extended, radiate from one point, i. e., if the seats are conical then the apex of the cone forming the upper seat and the apex of the cone forming the lower seat are one common point; or, if one of the seats is flat and the other conical, then the apex of the cone is a point on the surface of the flat seat. It is plain that by this mode of construction any trouble due to difference of expansion in the valve and seat is eliminated for the simple reason that the distance between the apex of the seat cones is equal to zero. If this distance is of any length whatsoever then this length will expand differently in the seat and in the valve.

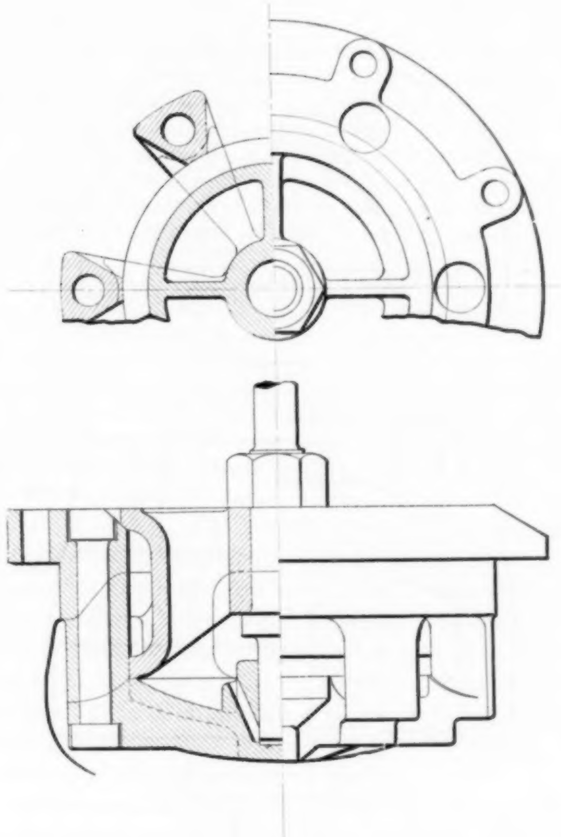


Fig. 1 Double-beat Poppet Valve for Steam

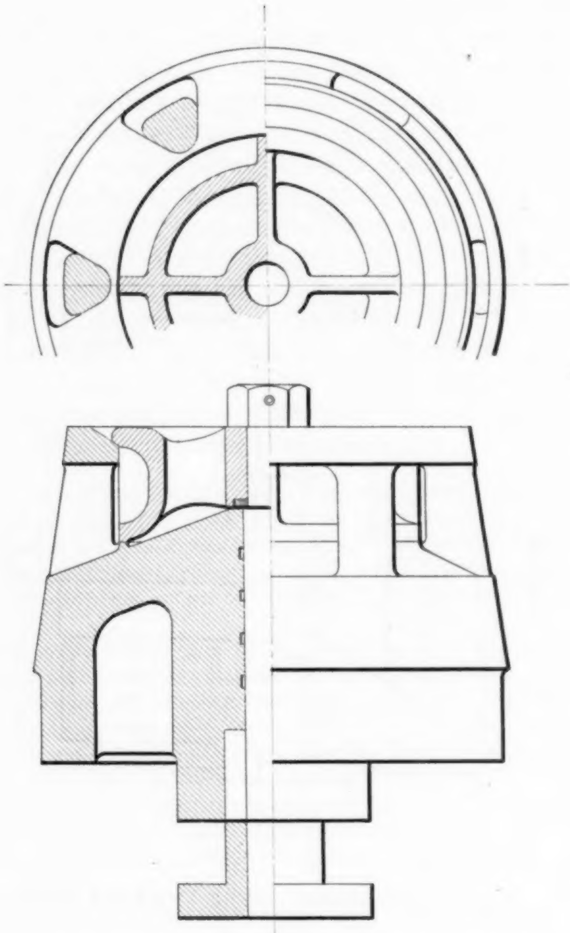


Fig. 2 Double-beat Poppet Valve for Exhaust

7 My experience is that it is of much greater importance to provide against expansion by pressure and if this is attended to, the valve and seat made of proper material, and the valve otherwise of good design, it is easy to make it tight and it will keep so indefinitely. One of the greatest mistakes in trying to make such valves tight is doing too much grinding. The valves should be gotten down to their seats when they are turned and by scraping and only the very last fitting should be done by grinding.

8 It is also a mistake to try to balance such valves too perfectly. If that is done there will be trouble in keeping them tight. The upper seat must be larger than the lower one by a liberal amount in order to keep it down on its bearing.

9 I believe that a great deal of the talk about poppet valves being hard to keep tight refers to the old style valves such as were used on the beam engines on paddle-wheel steamers. I have seen a great many designs in which it would appear that there was great danger of distortion of the valve seats due to pressure and heat in the cylinder, and this matter should be carefully guarded against in any poppet valve.

10 When I was in Europe a year ago Sulzer Brothers showed me a number of very large poppet valves made with *four* seats. They say it is no more trouble to make these valves tight than the ordinary style of poppet valve.

11 With reference to the valves shown in the illustrations, these valves have now been running a good many years and they are absolutely tight. The seats are as bright as mirrors, without the slightest indication of cutting or leaking, and when the valves are tripped and steam of 250 lb. pressure is turned on the inlet valves no steam comes out through the open indicator cocks of the cylinder.

Dr. W. F. M. Goss The results presented by this paper are likely to be received with some surprise by those who have not followed the development of the high speed engine with more than ordinary care. But as a matter of fact, it is probably true that the results reported are not especially exceptional. It would be wrong however to draw from them a sweeping conclusion to the effect that the more highly articulated type of engine is no better than the simpler type. The paper presents a fragment of the story only; the rest of it should be known before broad interpretations are attempted.

2 The conclusion of the paper, that while the more highly articulated types of engines are capable of rendering more efficient service than those of simpler type, they do not do so after being for a time in rough and tumble of service, is not so much an argument against the more highly articulated types, it seems to me, as it is an argument favoring greater attention to the details of their maintenance. The whole question is one which is very common and very old. In most classes of machinery, higher efficiency calls for greater refinement in design and greater refinement in design demands additional attention, and failure to bestow this additional attention often defeats the whole purpose of the designer.

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National Association of Railway Commissioners. *Proceedings.* 1898, 1899, 1900 and 1902. 4 vols. *Washington.* 8vo, cloth.

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American Railway Engineering and Maintenance-of-way Association. Vols. 4, 5, 6, 7 and 8. Completing the set of these Proceedings in The American Society of Mechanical Engineers Library.

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Concrete Structural Engineering. By E. Godfrey. *D. Van Nostrand Company, New York,* 1908. 16mo, leather, 448 pp., $2.50 net.

Contents, by chapter headings: A Survey of the field of Concrete Design; Cement; Lime; Aggregates; Mortar; Concrete; Reinforced Concrete; Handling and Placing Concrete; Setting and Hardening of Concrete; Finishing Concrete Surfaces; Forms for Concrete Work; Properties of Concrete; Notes on General Design and Construction; Estimating Cost; Specifications; Design of Reinforced Concrete Columns and Footings; Design of Reinforced Concrete Retaining Walls; Reinforced Concrete in the Making; Design of Reinforced Concrete Arches; Design of Foundations; Design of Reinforced Concrete Columns.

The Scientific American. Vol. 11–14, 1855–1858. Completing the set of the publication in The American Society of Mechanical Engineers Library with the exception of vol. 1 (1845).

Elements of Railroad Engineering. By Wm. G. Raymond, C.E., LL.D. *John Wiley and Sons, New York,* 1908. 8vo, cloth, 405 pp., $3.50.

Contents, by chapter headings: Introduction; Permanent Way; Alignment; Rails; Rail Fastenings; Cross Ties; Ballast and Road Bed; Culverts, Bridges and Minor Structures; Turnouts; Side Tracks and Yards; Elevation of the Outer Rail; Signaling; The Locomotive and its Work; Locomotive and Grade Problems; Railroad Expenditures; Effect on Operating Expense, of Change in the Number of Trains, the Tonnage Remaining Constant; Discussion of the Effect of Distance, Rise and Fall, and Curvature on Train Mile Costs; Problems in Change of Ruling Grade, Distance, Rise and Fall, and Curvature; Railroad Location, Construction and Betterment Surveys.

EMPLOYMENT BULLETIN

The Society has always considered it a special obligation and pleasant duty to be the medium of securing better positions for its members. The Secretary gives this his personal attention and is most anxious to receive requests both for positions and for men available. Notices are not repeated except upon special request. Copy for notices in this Bulletin should be received before the 15th of the month. The list of men available is made up of members of the Society and these are on file, with the names of other good men not members of the Society, who are capable of filling responsible positions. Information will be sent upon application.

POSITIONS AVAILABLE

016 Technical men and engineers to offer standard waterproofing compound for cement brick or stone walls. Liberal commission and exclusive territory to men who produce results.

017 A casualty insurance company invites applications for vacancy in inspection department, men having extensive experience in erection and operation of boilers, elevators and machinery in general. Full details of experience and references should be submitted with all applications.

018 An experienced designer of automatic machinery having practical knowledge of economic production. Location Western New York.

019 Instructor in mechanical engineering, classes in steam engines, thermodynamics, mechanical engineering of power plants, to handle several sections in experimental engineering laboratory, covering work in testing of steam and gas engines, fuels, etc. Salary $1200.

020 Position available for draftsman and salesman. One acquainted with pneumatic and hydraulic machinery. Location Chicago.

MEN AVAILABLE

105 Junior, mechanical engineer, technical graduate, desires connection with firm building commercial motor cars or stationary gas engines. Experienced in motor car designing and construction.

106 Member, technical graduate, with twenty years experience in design, construction and maintenance of manufacturing plants, mill construction, reinforced concrete, etc.

107 Graduate, mechanical engineering, Purdue University. Two years practical shop and construction work, drafting experience, mostly power plant, heating and ventilating, conveyor, electric railway, and general industrial and buildings

work, commercial testing. Past three years, engaged in industrial engineering. Would like to become identified with some good industrial concern and would expect $2500 to $3000 a year to start.

108 Electrical and mechanical engineer, conversant in power station and electric property construction and development, production and distribution of electric energy. Technical graduate. Affiliated with technical societies. Desires to make new connections.

109 Draftsman and assistant master mechanic of six years experience building construction and machine design, desires a change of work such as superintendent of construction or master mechanic.

110 Graduate electrical engineering, Massachusetts Institute of Technology; experience leading consulting and contracting engineering companies; large street railway company. Office and field designs, calculations, construction, tests and operation of steam electrical and hydraulic plants, apparatus and railway equipment.

111 Engineer to manage, develop, or improve a mechanical business, machine or process.

112 Practical and theoretical mechanical engineer of wide experience, boilers, power plants, gas engines, conveying machinery, and selling with special information on problems concerning the transfer of heat, desires responsible position; has had experience and will consider professorship.

113 Mechanical engineer, assistant superintendent or assistant manager, with large firm manufacturing gas, gasoline engines or motor trucks. Four years experience, rising to superintendent. Technical education, executive ability, systematic, energetic. Would consider also office manager with consulting mechanical engineer.

114 Member, 31 years of age, graduate of leading engineering college, past eight years designing, in charge of construction or both, mill buildings, power plants, special machinery, etc. Would consider position teaching any of these subjects.

115 Member, experience in machine construction, combustion motor operation, and in charge of installing and initial operation of several electric railway and lighting plants. Open for temporary or permanent engagement.

116 Member, located in Chicago, open for engagement. Wide experience in construction, operation and maintenance of industrial plants. Has been responsible head for design and installation of large power and refrigerating plants, and for working out plans for the economical and rapid handling of all kinds of material. Technical training, familiar with cost of material and production.

117 Position wanted as manager or superintendent, 15 years experience as superintendent on high-grade work. Expert on interchangeable work, also on most approved machine shop, foundry and boiler shop methods. Good organizer, can handle men and machinery to the best advantage.

118 Member, desiring change, designer, constructor, supervisor, can investigate any problem, and if of frequent nature, can reduce same to routine practice; experience power plants, mill work, brass, cement, blast furnace, concrete, structural, surveying. Thorough and systematic.

119 Junior, graduate mechanical engineer, Stevens Institute of Technology, eight years experience in machine and power plant design, desires to make a change. Prefers location in New York City district.

120 Mechanical engineer, graduate Stevens Institute of Technology, three years experience in the United States and six years in Europe, design, construction and operation electric railway and power plants; knows three languages, desires position with engineering firm.

121 Draftsman, representative salesman, or assistant superintendent, in general engineering lines. Location immaterial.

122 Junior member, age 28, desires position as superintendent or engineer of erection. Five years practical experience in design and construction of condensers, pumps, heaters, etc., erection of Corliss engines Two years designing and estimating pipe work for large power plants; shop, drafting room and business experience, executive ability.

123 Mechanical engineer, superintendent of manufacture or works or chief draftsman, American, practical and technical experience, good executive ability, thoroughly up-to-date automatic machine and machine tool designer for interchangeable manufacture. A-1 references. Desires to hear from companies, corporations, or persons in or about New York or Jersey City, who require the services of a first-class man.

124 Junior, graduate from Columbia, 3½ years designing, detailing and estimating reinforced concrete and fireproofing, one year with general contractor.

125 Member having 25 years experience, wants position of shop manager or general superintendent, competent to take entire charge of factory; expert in small interchangeable manufacture.

126 Member desires position on road as salesman for good firm.

127 Technical graduate, age 37, experience as consulting engineer and purchasing department, selling department and manager of a plant building engines, compressors and heavy machinery. Thorough business training, good organizer, familiar with details of foundry and machine shop management.

128 Assuredly and permanently located in Chicago with prominent firm of power plant specialists. Technically educated as mechanical engineer, broad experience. At present looking after sales. Will be interested in a proposition where the personal equation coupled with ability is a factor.

129 Technical graduate, age 33, nine years experience as draftsman, designer and chief draftsman; 2½ years chief draftsman with one of the leading concerns in the manufacture of air compressors, steam engines, rock drills, core drills and pumping machinery.

130 Member having 28 years experience, steam machinery, operating engineer of well known ability and reputation, patentee of various apparatus which has been commercially exploited, seven years with large industrial concern as steam and mechanical expert, recently conducting consulting office, will consider early engagement where high class work and economy are essential.

131 Junior, married, graduate mechanical engineer, experience in machine shop and stationary engineer (light and water plant). Four years building, testing, erecting and operating horizontal steam turbines. Also general power house construction work. Desires position where his experience and training will be of value.

132 Associate, age 31, desires to make a change. Twelve years varied drawing room experience, including checking and charge of men. Prefer work on heavy machinery and furnaces, rolling, steel and tube mills. Location preferably within 100 miles of New York or Philadelphia.

133 Young man, graduate mechanical engineering, thorough experience machinist and draftsman, having held executive positions in large manufacturing plants, desires position as superintendent or manager of a metal manufacturing plant or as an understudy to superintendent or manager who wishes to be relieved of some of the details.

CHANGES OF ADDRESS

ADAMS, Thomas D. (Junior, 1906), Southport, Conn.

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